AN INTERACTIVE KNOWLEDGE-BASED SYSTEM FOR HIGH-SPEED SPINDLE DESIGN

By

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The performance objectives of higher speeds and greater power utilization combined with increased accuracy of the machined surface have magnified the importance of the spindle design process and accentuated the need to develop and integrate computer-based techniques in design.

The structure and function of an interactive system for the process description, preliminary design, and evaluation of spindle-bearing systems for milling is described. Founded on a knowledge-based approach, an "optimized compromise" to the often conflicting spindle design objectives is presented in the form of a conceptual design. The design and analysis system provides a means by which the functional requirements of a spindle may be used to define performance objectives and design parameters in order to enhance the initial design process. Novice designers are guided in the design of a

reasonable, well-conceived spindle-bearing system utilizing existing spindle knowledge and empirical data. In addition, the spindle stiffness is optimized for the purpose of maintaining stable machining. The system provides on-line help and explanation capabilities in order to meaningfully describe the significance of influential process parameters affecting the spindle design and to explain the reasoning of system prompts.

Based on the process information, early specification of required product performance and underlying objectives may be utilized to reduce the overall design time as well as the potential of costly redesign. Through the incorporation of both the spindle form and function in the design process, sound decisions and evaluations of design alternatives may be made earlier and a reduction of trial and error iterations may be achieved. In this way, the performance specifications do not rank as merely collateral information for the design but are used to drive the design process.

Finally, a conceptual spindle design model may be effectively employed to predict the characteristic behavior of the final product despite the lack of detailed design information. A more complete evaluation of the conceptual design may significantly reduce the need for exhaustive verification of a spindle-bearing configuration later in the design process and allows for more timely feedback resulting in effective design modification.

INTRODUCTION

The future of machine tools dictates substantial improvements in accuracy and metal removal rates in machining; therefore, the modern machine tool industry is undergoing rapid change and development. Increasing productivity demands and the desire to exploit the potentials of machines through increased automation have resulted in greater requirements on machine tool spindles. The development of a new generation of machine tool spindles is guided by market demands, new technology, and machine design. The performance objectives of higher speeds and greater power utilization combined with increased accuracy of the machined surface have magnified the importance of the spindle design process and accentuated the need to develop and integrate computer-based techniques in design.

The performance of a machine tool is ultimately judged by the ability to produce a workpiece with required physical attributes at a minimum cost. Developments in tool materials and spindle design have led to the use of high-speed milling in order to increase metal removal rates and to correspondingly reduce machining times. As spindle speeds increase, the tooth impact frequency, defined by the product of the spindle speed in revolutions per second and the number

of teeth on the cutter, approaches or exceeds the critical natural frequency of the machining system. The excitation due to the tooth impact may significantly influence the dynamic response of the system leading to serious limits on the metal removal rate. It has been recognized that a decisive measure for the design of machine tool structures is the criterion of stability against chatter. The source of instability which imposes limitations on the milling process is self-excited vibration due to mode coupling and/or the regeneration of surface waviness. The performance of the machine tool is therefore directly related to the dynamic stiffness between the tool and workpiece.

The spindle-bearing system is a critical element in determining the machine tool performance because the spindle is invariably one of the most flexible components of the structure and directly affects the quality of the workpiece. The spindle fulfills a variety of functions including holding the tool, guiding the tool with adequate kinematic accuracy, transmitting drive torque, and absorbing external forces with minimum static, dynamic, and thermal distortions. A poorly designed spindle may result in excessive manufacturing costs due to the rejection of defective or out of tolerance parts, unacceptable surface quality requiring manual finishing, or by imposing the stability limitations resulting in low metal removal rates. The trend in machine tool development is clearly toward the design of very stiff spindles rotating at high speeds.

The art of modeling, computing, and designing spindles is highly developed. Numerous scientific papers addressing specialized aspects of machine tool design and performance exist. While clearly an immense database of knowledge has been accumulated, it would seem that only a fraction of this information and experience has been implemented in machine tool development and practice. Several possible reasons may be attributed to this condition:

- a lack of knowledge transfer due to an education gap or an increasingly aging workforce "retiring" knowledge;
- research results presented mainly in the form of published papers with restricted practical feasibility;
- an increased distinction between "research" and "development" activities within universities;
- a poor translation of this knowledge into hardware and software products.

The development of a knowledge-based interactive design system for the application description, conceptual design, analysis, and evaluation of spindle-bearing systems for milling is explained. Through the integration of design, structural analysis, and expert system programming techniques, a knowledge-based system founded on deep knowledge acquired through research and experience in the areas of high-speed spindles, bearings, and machining processes is realized. The resulting system described in this dissertation is not an

expert system per se, but it seeks to extend and enhance the capabilities of numeric tasks associated with the preliminary design process. A shallow coupling of numeric and symbolic programming techniques is employed as well as a design methodology emphasizing concept engineering.

Chapter 1 describes the fundamentals of software engineering in order to define the planning process, the definition of software and hardware requirements, and the design of the software. Such topics are important for the development of commercially viable products based on computer software.

Chapter 2 discusses design theory and methodologies as well as the application of computer-based aides for design and manufacturing. The fundamental procedure of the design process is characterized and the organization of software models associated to design tasks described.

Artificial intelligence and expert systems applied to engineering are summarized in Chapter 3. The terminology, system components, and functionality of knowledge-based systems are explained.

Chapter 4 confronts several key issues concerning the design of high-speed spindles and suggests areas of continued research.

A complete functional description of the Spindle Design System is covered in Chapter 5. The development of the individual modules and explanations of operations are summarized.

Chapter 6 illustrates some of the system capabilities and explains the results of several examples used to verify the design and analysis capabilities.

Finally, conclusions concerning the system development and high-speed spindle design are delineated, together with suggestions concerning future work.

CHAPTER 1

SOFTWARE ENGINEERING

Competitive demands on man's time have created the need to substantially increase work volume through greater productivity and personal efficiency. Today more than ever, practical solutions to everyday problems are found through the use of digital computers. Continuing advances in microprocessor technologies have unleashed the potential of computer hardware. The distinct divisions between the hardware classes of super, large, mini, and microcomputers are rapidly disappearing. This immense computing potential is harnessed by the logical element of software. The software is becoming increasingly sophisticated to meet the needs of advancing applications. The machine level code of early computers has grown into a multilevel spectrum of "high level" computer languages including FORTRAN, Pascal, and C, as well as specialized languages such as APL, PROLOG, and LISP.

In 1980 the United States spent in excess of $15 billion on commercial computer software. While the cost of software is growing at an annual rate exceeding 15%, the productivity of software implementors is increasing at less than 3% [21]. This staggering unbalance clearly illustrates the need to

establish an efficient engineering approach to the development of software. As Sommerville states [64], "we must apply existing knowledge derived from more basic subjects in exactly the same way as the mechanical, electrical, and civil engineer applies the basic sciences."

Software engineering represents a series of specific management controls, technical methods, tools, and review procedures for the planning, development, and maintenance of computer software which is explained in detail Freeman and Lewis [21]. The discipline is founded on key criteria which form a systematic strategy to translate a perceived need into a realized software solution. Figure 1.1 depicts a software development procedure exhibiting a well-defined methodology which is traceable among steps and provides clear, documented control. The High-Speed Spindle Design System was developed in the manner illustrated and is explained in detail.

1.1 Project Planning

Initial project planning results in the development of an overall software plan which defines the system objectives. The system functions and desired performance objectives are explicitly specified as well as hardware, software, and supplementary system elements. The software plan summarizes the scope, functionality, and limitations of the project.

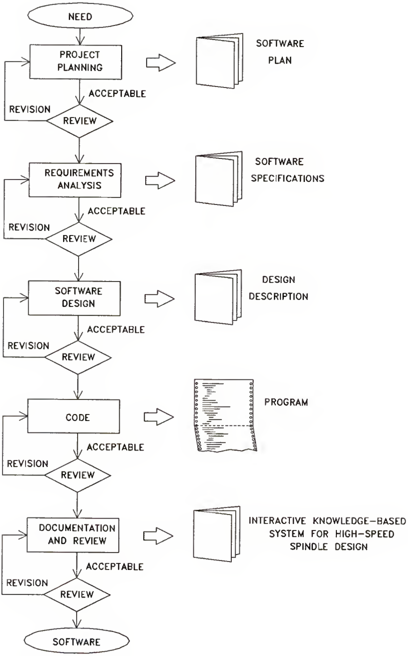


Figure 1.1 Software Development Procedure

The intent of the spindle design project is to provide a means by which the functional requirements of a spindle may be used to define performance objectives and design parameters in order to enhance the preliminary design process. The system directs novice designers and engineers in the design of a reasonable, well-conceived spindle-bearing system utilizing existing spindle design knowledge and empirical data. The spindle configuration is optimized with respect to the stiffness measured at the tool for the purpose of maintaining stable machining over the specified milling applications. In addition, the system is configured to provide standalone modeling and analysis capabilities for bearings and existing spindle-bearings concepts or design alternatives.

The software is written with strict adherence to the ANSI standard for FORTRAN 77 in order to provide the greatest portability among hardware platforms supplied with a FORTRAN compiler. The program is specifically designed for an IBM-compatible personal computer (PC), equipped with a graphics monitor. The recommended PC configuration includes 256+ KB of random access memory (RAM), a hard disk, a math coprocessor, an enhanced graphics adapter (EGA) with a compatible color monitor, and an Epson-compatible dot matrix printer. The graphics hardcopy is obtained by means of a small additional program which accompanies the design system and provides the utility for EGA screen dumps to the dot matrix printer.

A performance objective was set to limit the user's nonproductive waiting time by maximizing the system response and minimizing the required input key strokes. In order to assure that the program can be executed from a single 360 KB floppy diskette, the disk storage requirements have been reduced by limiting file storage.

1.2 Software Requirements

The definition of the software requirements provides a foundation for the software development and concludes with the program specifications. Software analysis tasks involve the study of information flow and structure requirements. An in-depth definition of the program functionality aids in determining software constraints and defining validation guidelines.

The project goal is to produce a spindle design system where a relatively novice designer or engineer will be interactively prompted through an application description of a desired spindle for the purpose of establishing a conceptual spindle design which meets the machining process requirements. In addition, the system may be used as an analysis tool. The system provides an on-line help and explanation capability in order to explain the reasoning of various system prompts and describe the significance of influential design parameters.

From the user input or assigned default values, the design requirements and critical design parameters are determined in order to parametrically configure a conceptual spindle design. An appropriate bearing configuration is chosen and the bearing stiffnesses are estimated based on design parameters and requirements or calculated based on the user input of specific bearing geometries and information. The bearing span and spindle overhang are statically optimized to minimize the spindle flexibility as measured at the tool. The interactively specified spindle design is automatically described for the purpose of analysis and evaluation or a user specified spindle description may be prepared interactively or off-line. The spindle-bearing system is modeled based on finite element modeling and the natural frequencies and characteristic mode shapes of the system are calculated. Both graphics and file utilities aid in the evaluation of the spindle configuration. Through the calculation of the direct transfer function at the end of the machine tool, the critical mode of vibration influencing machining stability is determined. The analysis of the spindle is evaluated and a diagnosis of possible sources of excessive flexibilities are presented and explained. The designer may interactively modify the spindle design to determine the influence of design changes on the system's behavior or modify the conceptual design to better meet application requirements.

1.3 Structured Design

The software requirements may be further specified by determining the program structure necessary to develop a computer program which is sophisticated enough to meet the project objectives but results in the most elementary structure possible. The goal of structured design is to create programs which are simpler and can be more easily understood, checked, programmed, debugged, and modified piece by piece [67]. The structured programming methods are important when building software systems to ensure the user's intended functionality is consistently and completely met. In addition, program adaptability is intrinsic to the system developed because of the design legibility, congruent structure, and self-containment.

The structure of the system information influences the final design of the software. Regardless of the application, the data are normally related through sequence or hierarchy. The standard data structures are illustrated in Figure 1.2. The scalar item, sequential vector, and N-dimensional space are relatively primitive in nature but may be effectively used for the data representation of the design project. These data structures correspond to a program variable, vector, and array, respectively, in the FORTRAN language. The linked list structure is commonly employed by programs written in languages which provide the support of system pointers, and may be used for the dynamic allocation of data storage space.

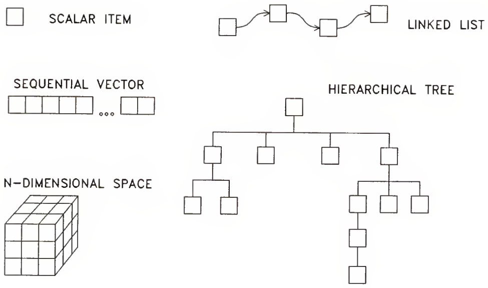


Figure 1.2 Standard Data Structures

The hierarchical tree provides the most intricate data structure. The branches of the tree imply a degree of associativity between the data, while the level of data detail increases with the depth of the structure.

The most appropriate structure for a particular system application is dependent on the operational characteristics of the program database. The size, design, potential for expansion, and data manipulation techniques, as well as the degree of associativity and generalization of the database, establish the fundamental requirements.

The primary trend in mechanical design systems is clearly toward object-oriented databases in a hierarchical structure. Two existing commercial program environments demonstrating

this structure are the Concept Modeler and ICAD (Intelligent Computer-Aided Design).

The scalar item, sequential vector, and 2-dimensional space structures are used within the Spindle Design System and may be exemplified by the following. A scalar item forms a single holding location for data. The single space is called a variable in FORTRAN and is generally used to store a numeric value referenced to a storage location by the address associated with the variable name. In the design program, variables are also used to store character data such as the linear distance unit name assigned by the system in response to the unit system chosen by the user,

```
CDIST = 'mm'   or   CDIST = 'in.'
```

Sequential vectors are used to store a string of data which are usually associated in a logical way and are referenced by the vector name and an index. The workpiece materials which are automatically handled within the design program are stored as character strings in a sequential vector format,

```
MATLS(1) = 'Aluminum'
MATLS(2) = 'Cast Iron'
MATLS(3) = 'Copper'
    .
    .
    .
MATLS(20)= ' '
```

Although the first 7 material types are explicitly specified by the system, additional storage space is assigned in order to accommodate user specified materials not listed by the program.

Finally, 2-dimensional structures or arrays are used to store data which are conveniently structured in a table format. In this example, rows of information corresponding to each machining application are stored for ranking and reference. The application number, NPT, is the common thread to the information within a row, while the same type of information is stored in a given column. Because in FORTRAN an array structure may be used for a single, predefined type of information, either integer, real, or character, the data may be represented by index codes. The material name used in a particular application is stored in array DATA by its index location, MCODE, within the character vector MATLS,

```
DATA(NPT,1) = NPT
DATA(NPT,2) = MCODE
     .
     .
     .
DATA(NPT,20)= FREQHZ .
```

Because of the inherent 64 KB memory stack limitation within the Disk Operating System, some "non-functional" common memory is utilized in order to accommodate the over 30 subroutines and modules. By defining larger data structures, such as arrays, as common data which are not used globally,

the memory requirements of the program may be divided between memory segments. The stack is a high memory segment which is used for saving the contents of function and subroutine calls and for the temporary storage of local variables. The memory segment below the stack is used to store all static data and common data which are not initialized. Such a programming technique may be utilized when developing a large Fortran program for use on a personal computer.

1.4 Software Design

Software design is the process through which the problem requirements are translated into a representation of software. The design does not represent procedural aspects such as the sequence of tasks, conditions under which the tasks are performed, looping, or cycles, but rather the architectural techniques used to construct the software system.

A hierarchical organization is utilized in order to fragment the process specification and preliminary design phase for the spindle design into a number of unique tasks of limited scope requiring only localized knowledge bases. The hierarchical organization exhibits technical qualities that make intelligent use of control and work providing programming flexibility through the implementation of modular software.

1.4.1 Software Modularity

Modularity is the single attribute of software that allows a program to be intellectually manageable. Each group of tasks which perform a single independent function is combined to form a system module. The software modules constitute the building blocks of the complete system. Each module is coded and debugged separately then integrated one at a time into the overall program. Increased modularity enhances the ability of detecting, isolating, and correcting design and coding errors and results in a more reliable program which is easier to maintain. Each module should strive for a single entrance and exit, a solitary function, and an aversion to dependent relationships with other modules.

As Figure 1.3 portrays, the varying cost of effort of the program development is related to the integration of the numerous software modules. Although the cost of effort per additional module decreases as the number of modules expands, potential problems arise with the increase in modules. As the number of modules multiplies, the interface and control complexity also increases therefore eliminating the diminishing cost of additional modules. The overall effective cost of the programming effort is displayed by the dashed curve and illustrates an optimum region of minimum cost. The desirable range of the number of software modules coincides with the hatched area.

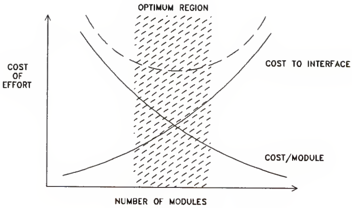


Figure 1.3 Software Development Costs

The technique used to integrate the numerous modules also influences the overall effort to develop the software system. The two principal methods of module integration, testing, and validation are the top-down and bottom-up approaches [64,81]. The top-down approach tests the major control flow of the program first and emphasizes the early integration of modules providing the ability to rapidly demonstrate the system functionality. A bottom-up integration procedure stresses the independent development and testing of modules. In this way, the isolation of errors is facilitated and module functionality may be displayed by drivers as well as standalone modules which are slightly modified during integration into the system. The bottom-up integration strategy was used to develop the modules independently and maintain maximum control flexibility. This modularity enables

the user to access the system in a variety of fashions depending on the level of knowledge and design detail defined. The bottom-up integration approach results in a single predominant system control module.

1.4.2 Module Independence

The qualitative measures of module independence are the cohesion and coupling of the software modules. In order to efficiently code, debug, and maintain the spindle program, a high level of cohesion is demonstrated while the level of module coupling is limited.

Cohesion refers to the degree to which a module performs a single isolated task. A series of terms are employed to relate the relationships between the different processing elements within a program module. The more closely related the elements, the easier the identification of the module's purpose and operation. Ideally, a module should consist of only elements required to perform a single, unique function. The terms used to describe the level of cohesion of a module follow. The highest degree of cohesion is the most desirable.

When little or no inherent relationship between module elements exists and the functions appear randomly divided, the cohesion is said to be coincidental [81]. A logical cohesion results when the module does not perform a single function but several logically related functions. Modules

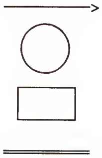
exhibiting logical cohesion are usually general in nature and require control elements to decide which function to perform. If the amount of cohesion is similar to the logical level but the elements of the module are related in time, such as at startup or as an initialization process, the cohesion is termed temporal. A module which consists of iterative decision making, loops, or the processing elements of an algorithm, demonstrates procedural cohesion. The Spindle Design System predominately illustrates a sequential cohesion. The system divides the preliminary design process into a series of steps that perform specific functions and distribute the operations over several modules. Output from one step in series becomes input data for the next. The highest measure of cohesion is represented when every processing element in the module is an integral and essential part accomplishing an exclusive task. The module is said to exhibit functional cohesion.

Whereas cohesion measures the relationship of elements internal to a module, coupling measures the strength of the connections between modules in a program. Module coupling is also referred to as binding as described in Stevens [67]. Although a program contains modules with varying degrees of coupling, a "looser" program exhibiting limited coupling is most desirable. A low degree of module coupling indicates a greater module independence and reduces the "ripple effect" resulting from the modification of a module. Content coupling is the highest level of coupling, established when one module

directly makes use of data or control information contained in another module. Content coupling frequently occurs when multiple entry points exist within a module and should be avoided. When a number of modules reference global or common data, the coupling is delineated as common. Common coupling may inadvertently affect the operation of other modules and complicates top-down integration. Ties to an external environment, library functions, or specific input/output devices are forms of external coupling. The graphics oriented modules in the design program are externally coupled to the hardware, although programming provisions were made to call the external device only when the system installation indicates it is appropriate. The passing of data and control information between modules is a measure of the control coupling. The control element affects the operation of the receiving module resulting in a moderate dependence on the calling module. The Interactive Design System primarily features the control coupling common to a bottom-up integration approach. When a minimum amount of communication between modules is necessary, the program illustrates data coupling. Data coupling is most desireable since only input and output data should flow across the connection between any two modules.

1.4.3 Information Flow

As information moves through the design program it is modified by a series of transformations. The spindle system exhibits data flow oriented design in which the information flow drives the software system and provides guidance to the software structure. One graphical technique for representing information flow within a program is a data flow diagram. The basic symbols and their associated meanings follow.



Data Flow

Processing and Transforming

Information Sources and Sinks

Available Information
(files and databases)

The flow of information among the modules and subroutines of the spindle design program is represented in Figure 1.4. The flow diagram emphasizes the need to maintain structured programming methods to effectively define, represent, and transform design data. Although the user is a direct source of information for nearly all of the modules, for clarification the flow diagram represents only interaction

between the user and the Design module. The Design module is the primary control module of the program. The descriptions and roles of each module will be explained in Chapter 5.

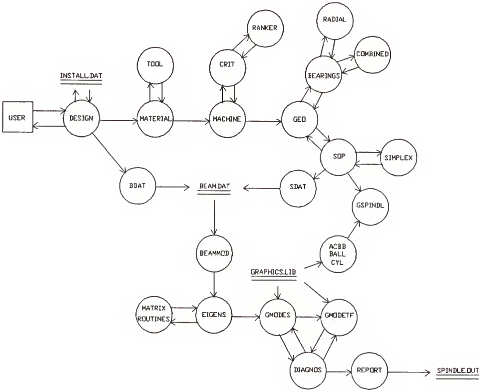


Figure 1.4 Spindle Design System Information Flow

In a hierarchical organization of software modules, the natural tendency is to allow the structure to fan out. The explosion results in a tree structure similar to the corresponding hierarchical tree illustrating the data structure in Figure 1.2. In structured programming the developer should strive for module implosion ensuing in a fan-in of the modules as depth increases [67]. The Spindle Design

System illustrates a contraction of modules which results in a main controlling root module, Design, and a single end branch, the Report module. Each of the system modules and subroutines shown in the information flow diagram for the design system will be explained in detail.

1.5 Software Interface

The software interface functionally ties the user to the program operation. The software interface is normally not covered in textbooks on software engineering but Sommerville dedicates an entire chapter to the user interface [64]. Off-line interfaces refer to the preparation of program readable input presented to the software system. On-line interfaces are interactive in nature and allow the user to directly communicate with the computer system. Both interface methods are utilized in the Spindle Design System.

Interactive styles of user software interfaces include computer-initiated dialogues and user-initiated dialogues. For most untrained users a menu-type or question-answer dialogue controlled by the computer is preferred. The computer presents a list of alternatives from which the user chooses or prompts the user through questions and takes action based on the replies. More advanced users prefer user-initiated dialogues usually either mnemonic based, employing a quasi-natural language, or form driven.

A simplified interactive user interface within the spindle program has been established to aid novice users. It is suggested that human memory is hierarchical in nature with short-term memory consisting of 7 or 8 memory locations [64]. In order to avoid information overloading, menus and limited question-answer dialogues are utilized. In addition to the menu choices, the user may enter a question mark (?) or request (H)elp when replying to a system prompt. The system will attempt to meaningfully explain the primary reason a question is asked and describe the affect the user's response may have concerning design parameters. If help is requested, default values or simple descriptions of the user's options are enumerated. The interface exhibits moderate fault tolerance and the system carefully checks user inputs.

Many commercial expert system shells have explanation facilities similar to RuleMaster 2 [56]. To gain insight into the system's line of reasoning or provide help to the user, proof-oriented explanations based on intent statements defined by the system developer are used. A fixed group of sentence masks form explanations of the system intent as the following examples portray. The developer's phrases are substituted within the arrows < >, and the present value of system variables are returned within brackets { }.

Since <intent> <transition verb> {returned value}
when <intent> <transition verb> {returned value}
and <intent> <transition verb> {returned value}
it follows that <intent> <transition verb> {returned value}.

An investigation of <intent> is being performed
in order to <verb> <intent>.

Although the sentence masks result in common forms of explanations, the help to the user may be limited because relatively simple explanations may result such as, "An investigation of <the groove radius> is being performed in order to <determine> <the bearing conformity>. The Spindle Design System typically provides more insight into the reasoning and affect of influential parameters as the following dialogue taken from the detailed bearing analysis subroutine called by the Bearings module demonstrates. The bearing conformity which is described by the explanation facility is represented in Figure 1.5.

```
Enter the radius of the race or groove (mm ) : ?

The radius of the race (groove) is used to calculate the
conformity between the race and ball.  The closer the race
conforms to the ball, the greater the frictional heat
generated within the contact.  Reduced conformity results
in increased contact stresses limiting bearing fatigue
life.  Most ball bearings have conformity ratios .51≤f≤.54,
with f = (radius of groove/ball diameter) =.52 being most
common.
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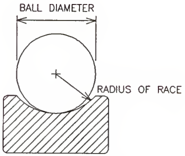

Figure 1.5 Bearing Conformity

1.6 Software Documentation

The software documentation is as much a part of the software system as the program code as it forms the critical element of the software configuration for the understanding, modification, and maintenance of the program. The program documentation consists of external descriptions and internal clarification. The documentation conveys the overall system objectives and the software's role within the system, as well as the major software functions, data descriptions, file structures, and communications. The internal documentation is embedded in the program code and provides information in addition to simply paraphrasing the code execution. The documentation within the program indicates the nature of any required modifications and a clear explanation of terms and procedures. This dissertation serves as the external system documentation while the program code is internally described by extensive commenting.

CHAPTER 2

ENGINEERING DESIGN

Engineering design is a creative process that is essential to the development and refinement of new products. The very nature of real engineering, manufacturing, and design problems have challenged the implementation of computer-based aides. These problems are often ill-posed, open-ended, and lack well-defined objectives. Because there are no standard criteria by which a design may be qualitatively measured, the design process generally lacks consistent procedures and guiding principles.

A variety of methods and approaches by which design alternatives may be evaluated have been developed. Mathematical and computer modeling, simulation, experimental prototyping and testing, along with the extrapolation of information from the past, have all been used to elevate the quality and efficiency of product design and manufacturing. Still, the design procedures of many products are significantly deficient of a systematic design methodology.

Recognizing the competitive need to further cultivate design research and education in the United States, the National Science Foundation sponsored a workshop addressing

design theory. The American Society of Mechanical Engineers (ASME) reported on the proceedings of the workshop held in Aspen, Colorado, in September of 1985, and published a review entitled, "Goals and Priorities for Research on Design Theory and Methodology."

2.1 Design Theory

In a summary of the complete ASME publication, Rabins et al. [51,p.25] define the principle areas comprising the discipline of design theory, as well as describe the terminology by the following,

> . . . design theory refers to systematic statements of principles and experimentally verified relationships that explain the design process and provide the fundamental understanding necessary to create a useful methodology for design.

The report summarizes the need for research and development in all of the areas of the design process. In conceptual design and innovation, the engineer identifies and defines the design requirements. The concept stage is the most creative yet least rational and understood step in the design process. The engineer must screen relatively abstract ideas in search of a feasible transformation to a physical configuration which may result in a successful final design. The report recognized the need to integrate other disciplines to

facilitate the understanding of the conceptual design process and the need to develop tools for the rapid evaluation of design concepts. The Spindle Design System integrates numerical programming with expert system features to aid in the automated assessment of an enhanced conceptual design.

The evaluation and measure of a design is achieved through quantitative and systematic methods. These methods are used to represent the design in an analytical manner which is amenable to numerical analysis and optimization. The procedures are employed to assess the design configuration's form, complexity, and manufacturability in order to accurately define a representation of the elusive design quality. The quantitative and systematic methods may be reinforced by a formalized design procedure and methodical evaluation procedures.

One of the most promising disciplines yielding a better understanding of the design process is Artificial Intelligence (AI). AI can be viewed as a form of programming that offers the unique features of heuristics, knowledge representation, and pattern directed invocation of program control. Intelligent and highly effective knowledge-based systems may facilitate the design process. The workshop called for the investigation of interactive design, the construction of knowledge-based interfaces to existing design tools, and the integration of multiple models, such as a structural and process model. Each of these areas of research are developed by the design and analysis system. Artificial intelligence

and knowledge-based systems will be explained in detail in Chapter 3.

Engineering design requires the utilization of vast and varied sources of information. The designer must be able to create, locate, use, and communicate the information efficiently and successfully. Preliminary design information usually consists of catalogue and library data, proprietary engineering standards, previous designs, and details from text and handbooks. The use and storage of design related information resources must be studied and methods for the effective presentation of the data must be developed. The Spindle Design System reports the current status of design parameters as well as performance data and portrays the information in numerical and graphical fashions.

The final area comprising the discipline of design theory is the investigation of the human interface. The interface includes the interpersonal communication among members of a design team and man-machine dialogue. It is important the engineering designers understand the tools which abet the design process and are able to employ them naturally and effectively. Research in the human interface aspects of design include expert assistants, the integration of multiple modes of interaction, and the characterization of information structure relationships.

2.2 Computer-Aided Design and Manufacturing

The use of Computer-Aided Design and Computer-Aided Manufacturing (CAD/CAM), systems has resulted in the evolution of the design process. The systems assist the designer by providing computer-based tools for generating new designs and modifying existing ones. Until recently, most CAD/CAM systems predominantly isolated the design process from manufacturing tasks therefore maintaining the "wall" between these activities which has existed for many years. Foreign competition and the loss of manufacturing industries in the U.S. has forced the integration of design and manufacturing to reduce costs and improve quality. Figure 2.1 illustrates the product design process in light of CAD/CAM depicted in [34].

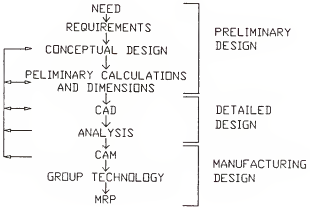

Figure 2.1 Design in Light of CAD/CAM

The preliminary design process is initiated by the recognition of a need. Through the definition of the product requirements the objectives of the design may be characterized. These objectives are formulated into a general concept of the solution required and give rise to the specification of viable design configurations. The feasibility of the configurations are explored in order to determine if the problem posed by the design project is solvable and to establish a set of potential solutions. The best application alternative from the conceivable solutions is used to form the conceptual design. Major design parameters are specified and the preliminary design is reviewed to confirm the potential of the concept. The defined conceptual design serves as a guide for the detailed design of the product.

During the detailed design stage the synthesis and complete specification of the components, subsystems, and assembly are described. The detailed design drawings are prepared and reviewed. The design is validated through the analysis of the complete system and the requirements of any design modifications are determined. The verification of the design is made by visually checking for clearances, calculations of tolerance stackups, determining area and mass properties, and the finite element modeling and analysis of the product.

Once the detailed design is finalized, the product model enters the manufacturing design stage. Through the use of the

CAM system, group technology, and manufacturing resource planning, the producability of the design is studied. The manufacturing process plan, design of tools, and necessary fixtures are defined. In addition, quality control parameters and standard times are specified and the product cost may be estimated.

2.3 Conceptual Design

During the design process multiple, even conflicting, requirements and constraints must be reconciled. Currently there is great interest in CAD/CAM and AI/Expert Systems both in engineering programs and at research institutions. Most of these activities focus primarily on decision theory [33], mathematical and structural optimization of the final design [30,31,46], and expert "assistants" which aid the user in selecting and executing complex analysis programs [1,14]. While these methods may prove to be successful, they are often highly time consuming and costly. As a result, contemporary CAD/CAM systems are most useful for the revision and documentation of nearly completed designs. The CAD/CAM models are mainly used as input to analysis routines and to drive CAM modules [77].

Typically the designer has little control or understanding of the process parameters influencing the design other than general overall objectives. In addition, the

critical tradeoffs, shown in Figure 2.2, between information, cost, and the ability to change the design are often overlooked [76]. In the earliest stages of the design process there is relatively little detailed product or performance information, although this lack of definition allows for the greatest design flexibility at the lowest cost. As the front end of product design, the conceptual stage possess the highest leverage on cost and performance. Most CAD/CAM and Expert System approaches require a relatively high level of product definition prior to the use of any automated methods. As a result, the feedback resulting in design changes occurs late in the design process when modifications are made under more constraints and at an increased cost.

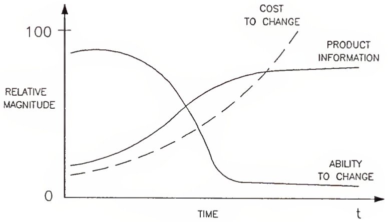

Figure 2.2 Cost of Design Change

Like most engineering problems, the solution space of possible designs is exceptionally large. The conceptual design phase is responsible for representing important attributes of a design and suppressing the myriad of irrelevant details [73]. Systematically exploring changes in key parameters is important for the development of understanding but is seldom performed within standard design procedures because it is time-consuming and tedious [76]. In conceptual design first-order engineering analysis techniques are practically employed to make use of the limited design knowledge.

Although the conceptual design stage frequently represents less than 15% of the overall design time, as much as 85% of the functionality, manufacturability, and cost are established. As Figure 2.3 illustrates, estimates suggest that 30% of the total product design time can be saved by developing performance objectives and verification data early in the design process. By reducing the "knowledge gap" which exists during the initial design stage, sound decisions and evaluations of alternatives to be made earlier [55]. In concept engineering the functional description of a device must be transformed into a detailed structural description. The transformation is dependent on the identification of the product purpose and intended use, the conceptual components and their interconnection, and finally the product form which is described by material specifications and the complete 3-dimensional geometry. The figure suggests that by defining

design objectives and increasing product performance knowledge in the concept phase of the design, influential design decisions may be made earlier and with greater certainty thus reducing the phase of product verification significantly. The use of integrated interactive computer graphics, analysis, and simulation during the development of the design concept can markedly improve the quality and efficiency of design and production. For these reasons and to more clearly define and explain the influence of design parameters, the Interactive Knowledge-Based System for High-Speed Spindle Design was developed.

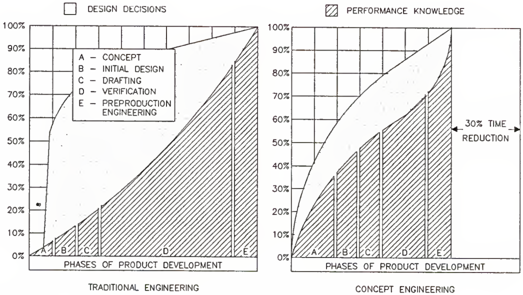

Figure 2.3 Concept Engineering [55]

During the design of machine tool spindles, only the general performance objectives of speed and power requirements are normally specified throughout the design process. The application of the spindle defined by the process information of machined parts is not indicated until well after the design and manufacturing of the spindle is completed, when the machine tool is designated by a CAM system. In the Spindle Design System, the engineering relationships are defined by "Analytically Deterministic" models of the milling process and spindle-bearing component behavior. Heuristics are additionally used to represent empirical data founded on speed, size, and lubrication information. "Rules of thumb" are employed to guide the selection of appropriate bearings and bearing configurations. In this way, the process information is integrated with the initial design phases of the spindle and realistic performance characteristics may be determined. The system provides a means by which the functional requirements of a spindle may be used to define performance objectives and design parameters to enhance the preliminary design process. The design system may also be applied to aid in the more complete specification of spindle criteria including structural characteristics prior to the purchase of a commercial spindle.

2.4 Parametric Design

A number of commercial firms such as Cognition and Iconnex offer conceptual design software that incorporates variational geometry. The combination of parametric and constraint-related design form a technique that can simultaneously solve a set of geometric constraints and equations. Governing equations that express engineering relationships among the elements of the design are employed to drive the procedure. Several of the commercial products that attempt to directly address conceptual design are reviewed in Villers [76].

The terms "parametric" and "featured-based" design are often used interchangeably. The terminology refers to a new generation of conceptual design tools which are compiled to construct systems that capture and convey the designer's intent within the design model [10]. Design attributes and configuration parameters are represented in a general fashion to define features. By simply changing the established parameters, the user may rapidly generate variations of a basic design concept in order to observe and understand the engineering relationships governing the design. In addition, a logical assembly of parts may be related through constraining associations which recognize the group as a functional unit. As a result, parametric systems can produce designs which are more meaningful than those generated by

traditional CAD/CAM systems which were initially automated drawing machines limited to drafting.

The original CAD/CAM systems replicated the manual process of design by providing an automated means of drawing the lines and curves on which geometric entities were founded. Today, 3-dimensional wireframe and solid modeling systems incorporate feature-based design to greatly enhance the design process. The systems typically offer a set of standard engineering shapes or features. Through the use of parametric programming, the designer may describe a complete design element such as a "blind, counter-sunk, threaded hole." While the designer describes the feature, the parametric system relates the graphical primitives, displays the design feature, and records the feature attributes. The feature-based systems allow both the designer and production engineers to communicate through a common language. By employing consistent terminology the same design model may be used from the conceptual design to the finished product. In the more advanced applications of design features, the systems allow production rules to be associated with the specified design elements. Recognizing machine and process capabilities, the user's design may be verified for manufacturability and a process plan generated automatically. For the previously described feature, the system would determine based on the finished hole diameter if a centerdrilling operation is required, specify the final drill size and position, the threading operation, and the countersinking.

In the most advanced feature-based CAD/CAM systems, additional rules concerning production, tolerances, and cost may be enforced. For example, a system may inform the designer that a "punched hole" positioned less than half the radius from an edge should not be specified in a sheet metal operation if the material thickness is over a critical value. Future systems may also fully incorporate machining process models with features to assure the quality of design.

The Spindle Design System does not explicitly exhibit the characteristics of the a featured-based CAD/CAM system, but it was conceived based on general parametric design systems under development. The designer's intentions and the design requirements are interpreted in the form of constraint descriptions. The Spindle System is a variation of the design task and product models which the AI-based, Man-Machine system at the University of Tokyo represents [33]. The Spindle System structure is illustrated in Figure 2.4.

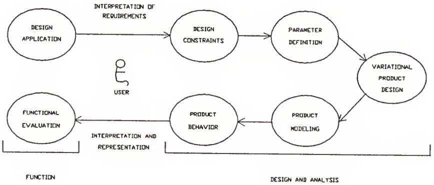

Figure 2.4 Interactive Knowledge-Based Design System

The Spindle Design System illustrates the definition of the product based on form and function. The interactive design system places the user at the center on the conceptual design process by aiding in the definition and representation of function requirements as well as providing a means of design evaluation. Throughout the design process the user maintains the role of decision-maker while the system provides updated information. The appropriate design proposal is made based on selected rules, catalogue data, computational formulas and the empirical data of past designs. The spindle-bearing configurations are parametrically modeled and the design concept is modified according to the formulated constraints and the designer's direction.

A program which performs domain independent parametric design by "iterative redesign" is Dominic II, described in detail in [46]. The program represents the combined efforts of the Mechanical Design Automation Laboratory at the University of Massachusetts and General Electric's AI branch. Dominic II models a class of parametric design which requires no conceptual innovation and incorporates optimizing algorithms. The system accepts the problem definition in more natural terms and seeks information about individual performance parameters. From the user's input, the problem parameters are defined within a specific domain and are controlled according design variable limits and an ordered priority list of the variables to be used to manipulate performance parameters. In a similar fashion to the Spindle

Design System, the Dominic II program employs several strategies to find "satisfactory" solutions rather than a design optimum.

Founded on a knowledge-based approach, an "optimized compromise" to the often conflicting spindle design objectives is presented by the Spindle System in the form of a preliminary design. Despite the knowledge gap which exists during the initial design stages, sound decisions and evaluations of alternatives may be made. A reduction of trial and error iterations may be achieved by the early definition of underlying objectives and constraints. A simplified model of a spindle is established during the conceptual design stage and may be used to adequately represent a spindle in order to predict the behavior of the realized product and reduce the potential of costly redesign.

2.5 Design Optimization

Optimized designs are generally defined as designs which meet the system requirements while minimizing other factors. The most frequently optimized characteristic is the structure of the system and therefore "design optimization" and "structural optimization" are often used interchangeably. Structural optimization is a numerical design technique for obtaining the optimum design of a system through the control of the design variables. A good design model requires a sound

mathematical formulation coupled with a degree of engineering judgement acquired through exposure to a range of problems and experience. Optimization is a technology which supplements an engineer's ideas and abilities.

Computer-Aided Design Optimization (CADO), shares the basic iterative nature and general procedure with the design process. As described in Elsaie [16], CADO utilizes the speed and accuracy of computer-based systems to analyze a design and make modifications of the preselected design variables within specified constraints in order to minimize the overall objective function. The performance objective of most systems is to minimize the cost, weight, stress, or flexibilities of a structure. Design optimization analysis available in commercial systems are listed in Rouse [54], along with a review of optimization research development.

Most design optimization processes follow an established procedure for developing an optimum design model. At the beginning of optimum design modeling, the designer must establish an efficient design model. For structural problems, the Finite Element Model (FEM), is frequently used to generate the optimum design model. Detailed design optimization of realistic structures may involve hundreds of design variables and implicit nonlinear constraints. Optimization and analysis is a necessary subtask of the design process and is used for judging the quality of the proposed design. Unfortunately, optimization of detailed and complex models employing Finite Element Analysis (FEA), are computationally

intensive and large models requiring 5 or more iterations may require up to an hour of design optimization time on a supercomputer [74].

The kind, number, and distribution of design variables must be identified after a type of design model has been specified. Typical design variables include cross-sectional properties such as the area, moment of inertia, thickness, and torsional rigidity, material properties, interfaces to external boundaries, and shape. Few successful algorithms for shape or geometric optimization are available. It is hoped that AI will eventually be employed to facilitate geometric optimization. Shape optimization requires the differentiation of structural matrices with respect to nodal coordinate vectors and has been successfully implemented in recent work at the General Motors Research Laboratory.

Once the design variables have been identified, expressions representing the constraints imposing restrictions on the design must be defined. Figure 2.5 depicts the design space defined by side and behavior constraints that are generally represented by a finite number of inequalities.

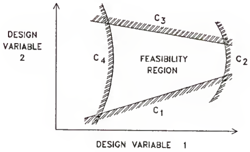

Figure 2.5 Design Space

The side constraints illustrated by C_1 - C_4 describe the lower and upper bounds limiting design variables and may consist of minimum and maximum values,

$x_i^l \leq x_i < x_i^u$ where x is a vector of design variables,

relationships of constraints expressed by governing equations,

$G_j(x) \leq 0$ where G_j (j=1 to n) are inequality constraints,
$H_k(x) = 0$ where H_k (j=1 to m) are equality constraints,

or specific behavior restrictions on the response of a structure designated by stress and displacement limitations, buckling, or natural vibration frequencies. The behavior constraints are derived from the performance requirements which are either explicitly considered or are formulated from the description of the system application as is the primary case in the Spindle Design System.

The most important properties of a design are represented by the system objective function. The function F(x), constitutes a basis for the selection of one or several acceptable design alternatives. The minimum of the objective function is sought by the optimization algorithm, although a maximum may be found by negating or inverting the function such as -F(x) or 1/F(x).

After the design model is established, the designer must determine a suitable optimization strategy and sensitivity analysis method for efficient optimization performance.

Optimization strategies include geometric programming, fully stressed design, optimality criteria method, and dual methods which are frequently used for optimization with discrete design variables. Sensitivity analysis methods include finite difference, design space, behavior space, and the virtual load technique. The choice of an appropriate method is dependent on the characteristics of the design model and may significantly influence the computational demands of the problem.

The first requirement of a good algorithm for engineering optimum design is reliability. The program reliability refers to the ability of the algorithm to locate at least a local optimum point regardless of the starting conditions. The second requirement for a robust algorithm is the efficiency of the program with respect to the computer run-time requirements. Because the criteria are somewhat opposing, no single algorithm meets both requirements in a general sense. Figure 2.6 classifies several of the mathematical programming techniques for nonlinear optimization. The mathematical theory of each algorithm is summarized in Fox [20] and Vanderplaats [75]. Each method has its own advantages and drawbacks, therefore the designer must choose the most suitable optimization strategy based on the design problem characteristics. In addition, the designer must generally select the user-defined program parameters consisting of the convergence criterion, active constraint strategy, push off factors, and step size. The values of the parameters can

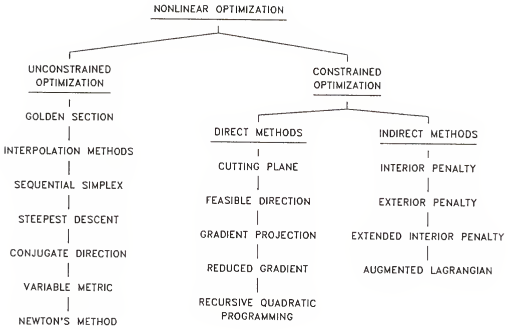

Figure 2.6 Optimizing Algorithms

significantly influence the optimization performance. The designer may be required to monitor, diagnose, and modify the algorithm parameters or even the original design model during the iterative optimization process. The searching technique for the optimum design is usually based on the following iterative equation:

$$x^q = x^{q-1} + a * S^q$$

where q corresponds to the iteration number, a is the step length, and S is the vector search direction.

Structural optimization has been applied to machine tools for the optimal dynamic response of the structure. In the area of spindle design, various design and optimization strategies have been explored to increase the static and dynamic stiffness of spindle-bearing systems in order to limit the effects of external force excitation and to avoid self-excited vibrations resulting from regenerative chatter. Design strategies have addressed several areas including the type, mounting, arrangement, and preload of bearings [68,71]; the static and dynamic determination of optimal bearing spacing and spindle overhang [60,17]; methods to dissipate vibrational response through increased damping; and the re-distribution of the frequency response of the spindle using tuned absorbers, selectively added masses, and the modification of the spindle geometry [53,66].

A relatively unique optimization approach for the dynamic modification of a precision grinding machine is presented in [43]. The structural optimization is based on a modal model. Although the relationship between the modal parameters and structural parameters is not direct, optimization may be performed in modal coordinates over a limited frequency band and with several simplifying assumptions. The optimum value of discrete structural members were determined by means of multi-factors optimization. The modification of structural models in modal coordinates have been investigated [66]. Although the modal model offers the advantage of a significantly smaller model even when the discrete, physical

model consists of numerous degrees of freedom, large errors may result when modifications of the original system are made and the original mode shapes change significantly.

The specific optimization strategy employed in the Spindle Design System will be explained during the module descriptions in Chapter 5.

CHAPTER 3

KNOWLEDGE-BASED SYSTEMS

Artificial Intelligence (AI), is a term which was first coined by John McCarthy at Stanford University in 1956. AI refers to a discipline within computer science which studies how machines might behave like people. Most practioners would agree that the primary goal of AI is to develop a computer which exhibits intelligent behavior. With such an objective, one is inevitably forced to question the very nature of intelligence. As suggested by the difficulty exhibited in describing AI, [11,58], the issue of defining intelligent behavior will be debated for many years to come. Despite the fact the logical deductions representing reasoning can be carried out by computers as pure manipulations of symbols without regard to content, intelligence and logic are often related. Intelligence is generally associated with a reasoning process requiring knowledge, therefore the secondary goals of AI are the representation and utilization of knowledge.

AI techniques offer an active medium in which to work with knowledge. Most applications fall into the two broad categories of problem solving or decision making based on

symbolic information, and intelligent communication systems. Several technologies popularly affiliated with artificial intelligence are listed below:

- image or pattern recognition
- speech recognition and generation
- cognitive modeling
- intelligent robotics
- expert systems.

Specialized languages have been employed for AI programming applications. In the United States, the list processing language of LISP was developed in the early 1960's and is surprisingly the second oldest high-level language following only FORTRAN in age. Founded on programmable logic, PROLOG is a popular AI language in Europe and Japan. The Japanese demonstrated a commitment to PROLOG in their famous Fifth Generation computer project, although PROLOG systems are at least a decade behind LISP in maturity [52]. FOURTH is an alternative language which has been increasingly utilized especially in real-time expert system applications. Other generally used high-level languages such as FORTRAN, PASCAL, C, and BASIC have been employed for AI applications. In fact, many commercial AI programs although developed in a specialized programming language, have been translated into C in order to maintain portability and to avoid the need for specialized hardware to increase performance.

3.1 Expert Systems

One of the most rapidly growing branches of artificial intelligence is knowledge-based expert systems. A review of AI activities within computer companies indicates an attractive business outlook and continued commercial growth. Over the last five years the expert system market has flourished. In 1985, sales of expert system software amounted to $74 million and specialized hardware sales grossed $300 million. By 1990, these sales figures are expected grow to $810 million and $1.7 billion, respectively [9].

Similar to the advent of robotics, the introduction of expert systems to the commercial marketplace was preceded by much hype and misunderstanding resulting in initial expectations which were not realistic. Expert systems are not a panacea for all areas of engineering which are poorly understood. On the contrary, their application should be in clearly defined, restricted areas where expertise exists. Expert systems may be more akin to idiot savants than to real human experts. Perhaps the most critical reminder concerning the application of expert systems is summarized [11,p.20] when Denning quotes Alan Perlis of Yale, "Good work in AI concerns the automation of things we know how to do, not the automation of things we would like to know how to do."

Expert systems are generally described as computer programs that capture specialized knowledge in a narrow and

well-defined domain. The systems simulate the reasoning process of a human expert in order to provide knowledgeable advice about a task. Dym [15] has surveyed expert system approaches applied to Computer-Aided Engineering (CAE), in great detail. The survey acts as an excellent tutorial of AI and expert systems as well as assesses the roles that expert systems can play in engineering analysis, design, planning, and education.

Edward Feigenbaum who is recognized as the father of the field of expert systems, states that an expert system consists of two main components, a knowledgeable user, and the expert system program [63]. Expertise demonstrated by the system includes knowledge about a particular domain, understanding the involved problems, and the skill for solving the problems. The generic knowledge-based system may be represented by the user, user interface, logic element, and computational element which are described by Milacic [44]. Figure 3.1 illustrates the block structure of a knowledge-based system as well as the logical organization of the Spindle Design System. The user is prompted interactively by the design and analysis system. Within the computational element of the system, the application descriptions of the spindle are interpreted in order to establish the performance objectives and constraints which form the application database. A simplified machining process model is used as the application program while the program modules contain localized rule databases. Represented by the logical element, the system knowledge-base contains

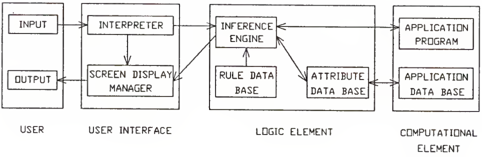

Figure 3.1 Structure of an Expert System

both factual knowledge about the spindle design in the form of production rules, and the control regime defining the method knowledge which manages the design process.

3.1.1 Knowledge Base

Expert systems are limited by the information in their databases and by the nature of the process by which the information is stored. The knowledge base contains domain facts which embody knowledge and understanding founded on basic principles, as well as heuristic knowledge developed from the application of the basic principles for a specific class of problems. Based on the level of domain understanding, knowledge-based systems may be described as deep systems if the conclusions are derived for models of phenomena in their domains using first principles embedded in

the models, or shallow systems which are generally designed for speed and store more facts than rules in their database. Shallow systems may also contain heuristics, or "rules-of-thumb," which represent experimental knowledge that is developed from successfully solving many problems in a specific domain. It should be noted that systems based on heuristic programming techniques and rule evaluation do not encompass all the systems which simulate human expertise. Although autopilots are expert fliers, they are normally not considered expert systems.

3.1.2 Knowledge Representation

A knowledge representation scheme and reasoning technique which is suitable for the addressed problem domain is critical to the effective application of an expert system. Several widely used knowledge representation schemes are employed by expert systems. The most popular representation method is the rule-based scheme. Knowledge is captured by a series of production rules which are formed by an IF (condition)-THEN (action) construction. Rules which contain knowledge about the use of other rules are called "meta rules." The concept and formulation of production rules is based on the human cognitive process. Rule-based representations are used most frequently because they offer a natural means for describing complex knowledge and are analogous to the familiar IF-THEN

control block in high-level programming languages. Although production rules simplify the description of a problem, contradictions may be introduced by the incomplete definition of a problem as well as by overconstraining conditions. An example of a production rule is illustrated below.

IF the spindle is dedicated to one specific application
THEN the spindle is a SINGLE_PURPOSE_SPINDLE.

A frame-based network data structure is used to represent the relations between concepts, objects, or events, and their attributes. Figure 3.2 depicts the conceptual representation of a semantic network and a frame. Both representations consist of a node and the links which correspond to interrelations. The networks typically incorporate the mechanism of inheritance lattices described by Eschmann et al. [18]. The frame-based system is centered on a hierarchy of descriptions of objects referred to in the system rules.

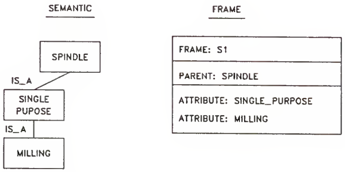

Figure 3.2 Network Data Structures

Logic-based knowledge representation schemes are founded on first order predicate calculus. As a subset of mathematical logic, predicate calculus is a formal language of symbol structures that can be used in computer applications to represent known facts and relationships subject to inferential reasoning. Most expert systems utilizing a logic-based representation method are developed using PROLOG. The basic units employed for depicting facts and rules are predications, which are presented by a predicate followed by a list of arguments. The following examples respectively illustrate the representation of a fact and the general form of a rule in a logic-based system,

```
IS_A ( single_purpose, spindle )

consequent : [antecedent_1,..., antecedent_N].
```

3.1.3 Knowledge Acquisition

Prior to the development of a knowledge base, a greater understanding of the problem and of the knowledge required to complete necessary tasks must be acquired. Currently, the knowledge acquisition process is performed by a "knowledge engineer" who is responsible for building the expert system. The difficulty in capturing the domain knowledge is that rather than defining algorithmic approaches to problem

solving, the knowledge to be acquired is often heuristic, judgmental, subjective, and not always consciously understood by the expert. The knowledge acquisition process may be divided into five stages:

- Identification
- Conceptualization
- Formalization
- Implementation
- Testing.

Identification refers to the characterizing of the important aspects of the problem by identifying the limits of the problem domain, knowledge sources, and system goals. The identification stage results in the specification of the problem requirements. Through the conceptualization stage, key properties and relations described by the domain-specific attributes are explicitly distinguished. Representation schemes and expert system tools are considered. Formalization is the stage in which a formal model of the task and its key properties are mapped into a representation scheme. The structure for organizing the knowledge is established. During implementation, the rules which embody the knowledge are formulated resulting in a prototype system. The complete expert system is an expansion of the prototype which has been thoroughly tested. The testing stage includes the comparison of application cases which are utilized in order to verify the

expert system and concludes with improvements, reformulations, redesign, and system refinements. The knowledge acquisition process was not only used for the development of the Spindle Design System but it is employed to establish application knowledge, performance objectives, and constraints during the spindle design process.

3.1.4 Inference Engine

Within an expert system there are three central, enabling concepts embodied by symbolic knowledge representation and symbol structures, a strategy for manipulating symbols, and search techniques for reducing the combinatorics in searches of the solution space. The inference engine is the means by which the problem solving process is controlled. In addition to solving the problem, the inference engine may produce an explanation of the solution. This ability to explain the path of reasoning is most frequently associated with convincing expertise. Computer scientists stress that the chain of reasoning must exhibit the quality of transparency while leaving an audit trail so that the user may explore the path to the solution. The choice of the inference engine is strongly coupled to the nature of the task the system is designed to perform.

Because design tasks involve limited data with a vast possible solution space, the Spindle Design System utilizes

a search technique for spindle design based on a generate-and-test methodology. Systems which proceed forward from the given data to the problem conclusions are said to be demonstrating a forward-chaining inference procedure. Forward-chaining uses a rule base to reason from the input data to problem conclusions and is therefore considered a data-driven procedure based on a protocol viewed as antecedent reasoning. The evaluation mechanism proceeds by tracing rules from left to right, until the ultimate consequences of the given conditions have been found. The process is analogous to the pattern of reduction which forms the basis of the inference algorithm for PROLOG, called resolution.

Most expert systems are based on an inference technique called backward-chaining. The backward-chaining process starts from a hypothesis or goal and checks if the hypothesis can be supported by available facts. The nature of this inference procedure is similar to the diagnostic process exhibited by a doctor. The backward-chaining strategy requires selecting a goal and then scanning the rules to find those whose consequent actions will achieve that goal. Because evaluation proceeds from right to left until the antecedents of a hypothesis about the given state have been found, the procedure is said to be goal-driven and demonstrates consequent reasoning.

Inference engines which combine both reasoning techniques to yield conclusions are based on mixed reasoning. In a predominantly forward chaining system, the rule conditions

which are satisfied at a state are collected. If more than one rule is applicable, a conflict-resolution strategy may be triggered to eliminate the alternative choices. A mixed reasoning procedure may employ a pruning technique based on backward-chaining in which conditions are inferred from partial goals. The mixed reasoning strategy may be most effective in problem domains where forward-chaining alone generates conclusions not directly related to the problem and a purely backward-chaining process exhibits difficulty shifting to alternate hypotheses after becoming fixed on an initial set of unsupported facts.

In addition to the reasoning techniques described, many inference engines can handle incomplete knowledge by associating confidence levels to knowledge and rules. The highest certainty level of 1.0 is assigned to facts, where unexplained information is weighed by a certainty factor approaching 0.0. The confidence level of the most applicable state is determined by the following,

new_conf_level=conf_level_1+[(1.0-conf_level_1)*conf_level_2].

The resulting conclusion from the inference engine is the state with the highest confidence level associated with it. The system continues searching that path until cycling occurs, one of the alternatives possesses a greater certainty value, or a goal state is achieved.

3.1.5 Input/Output Facility

The Input/Output (I/O) facility constitutes the element of the expert system responsible for the user interface. The I/O facility must interpret user input and manager the system output. The facility may contain fault tolerance, error checking of inputs, and natural language interpretation. The I/O facility directs the screen display and explanation routine as well as all connecting peripherals.

The I/O facility allows the user to communicate with the expert system and to create and use the database. Although most expert systems are lacking a means of knowledge acquisition within the I/O facility, an intelligent editing program, induction system, or advanced example interpretation and rule generator, may be developed to assume the functions of the knowledge engineer. A fully integrated knowledge acquisition and I/O facility will mark the transition to a higher level of automation in designing expert systems.

3.2 Expert System Tools

Knowledge engineering tools which are used for capturing knowledge and building expert systems may incorporate three elements requiring varying levels of development. The tools may be generally categorized into divisions including

programming languages, programming environments, and skeleton expert systems called shells.

Building an expert system by means of a programming language requires the highest level of programming ability and understanding of AI techniques. Specialized languages such as LISP, PROLOG, and the myriad of related dialects, may be employed as well as general programming languages such as FORTRAN, PASCAL, and C. The current trend in AI programming is to choose a conventional language, usually C, to deliver expert systems. The conventional languages reduce operational times and limit hardware requirements.

Programming environments are software environments created to facilitate the development and prototyping of expert systems. The development environment may vary in sophistication providing programming paradigms such as in the LOOPS environment, or a toolbox of high-level programming aids like those available in KEE or ICAD. The expert system environments may demonstrate two major themes combining programming models with the integration of an environment for development, debugging, and testing of the knowledge-based system.

Four programming paradigms which are frequently utilized within a programming environment are procedure-oriented, object-oriented, access-oriented, and rule-oriented programming. Procedure-oriented programming is similar to most familiar programming languages which separate data from programs and constructs large programs from smaller

subroutines. In an object-oriented programming model, information is organized in terms of objects which combine both data and methods. Objects are commonly said to know about themselves, how they are treated, and how they relate to other objects, therefore very complex objects may be defined in terms of simpler ones. Access-oriented programming is used in programs that monitor other programs. Active values are associated to objects that can trigger other computations when an object's data are accessed. These "objects in waiting" are called Demons and may be somewhat compared to interrupt-driven programs within conventional programming languages. Finally, rule-oriented programming is used to represent decision-making knowledge in the expert system and provides the mechanism for explanation and belief revision.

The most common commercial expert system development tool is the expert system shell. Shells are the skeleton of an expert systems which contain a domain independent inference engine, an empty knowledge base, and a user-friendly interface. With the use of expert system shells, the knowledge engineer can concentrate on knowledge representation and neglect the complex inference engine strategies. Simplified shells allow the domain expert to develop and test an expert system without the assistance of a knowledge engineer. A range of development tools enable the user to build a knowledge base for a specific application. The program aids within expert system shells may differ

significantly as do the shell costs which vary from $100 to in excess of $100,000. Several expert system shells have been summarized and evaluated in [23,40]. The application requirements of an expert system must be carefully specified before the purchase and use of an expert system shell, because a shell may lack provisions of typical engineering software facilities such as the following list summarizes [32]:

- floating-point and double-precision arithmetic operations
- transcendental and other scientific functions
- ability to call libraries and subroutines
- ability to work inside or with other programming environments
- general file handling capabilities
- ability to communicate with external databases
- facility to call operating system commands
- graphics capability with operator interaction
- communication with external peripherals
- ability to link with programs written in other languages
- the production of fast, compact code which is compilable.

The number of development tools for AI applications and the construction of expert systems is expanding significantly. Table 3.1 illustrates many of the characteristics associated with AI programs as well as the tools and methods used to develop knowledge-based systems.

Table 3.1 Expert System Development Tools

AI Programs	Knowledge Representation
- Representation - Decoding - Control of Search - Indexing - Prediction and Recovery - Dynamic Modification - Generalization - Curiosity - Creativity	- Rules - Frames - Inheritance - Uncertainty - Hypothetical Reasoning - Meta-Knowledge - Knowledge Acquisition Aids - Self-Learning Knowledge Base
Development Environment	**Inference and Control**
- Help Menus - Saved Cases - Trace and Probes - Breakpoints - Performance Measurements - Graphics Interface and Tool Kit	- Forward Chaining - Backward Chaining - Mixed Strategy - Demons - Truth Maintenance - Meta-Control

One software tool for the development of expert systems is RuleMaster 2. The program is a combination of a programming environment and shell which provides greater flexibility for the development of expert system applications.

A framework file is defined by the system developer in order to specify the structure of the module-based expert system. Induction files are used to form the knowledge base and inferencing mechanism. The program accommodates the representation of both declarative and procedural knowledge. Declarative knowledge describes the condition of a domain state by presenting facts and conclusions which are drawn from an analysis of the condition attributes. The RuleMaster program can automatically generate rules from provided example sets. Procedural knowledge describes how a process or task is achieved by providing ordered, step-by-step instructions. The developer may imply knowledge through example cases or explicitly define principles and procedures by programming in the Radial language within RuleMaster.

RuleMaster exhibits a modular, hierarchical structure which assists in expert system design, development, and maintenance consistent with sound software engineering principles. A variety of the program attributes were used as models in the development in the Spindle Design System. RuleMaster can generate expert systems in FORTRAN and C source code, therefore it can interface with other program applications or embed systems within the expert system environment. RuleMaster was used to induce rules from case examples and generate program code in order to provide a rule programming methodology in the Spindle Design System. Rules generated by RuleMaster are expressed as a hierarchy of decision trees. English-like explanation facilities are

provided by means of sentence masks discussed in Section 1.5. The screen display package within the expert system shell provides only a scrolling text display for FORTRAN programs. With the aid of specific run-time environments, RuleMaster 2 programs may be executed within the MS-DOS, XENIX, UNIX, and VMS operating systems. The RuleMaster 2 program was not used to develop the Spindle Design System for several reasons. The run-time environments are fairly costly and limit the portability of the application. Programming in the Radial language requires a similar level of work as programming in FORTRAN, and the resulting generated source programs are extremely difficult to understand. For an application as large as the spindle design process, the source code generated would have to be modified prior to compilation. RuleMaster is a valuable tool for the development of expert systems but was used mainly as a model providing guidance for the structure of the spindle program.

3.3 Symbolic and Numeric Computing

Any system linking both numeric and symbolic computing processes may be considered a coupled system. Numeric computing is characterized by the processing and reduction of data for the purpose of simulations and quantitative models based on numerical algorithms. In comparison to the accurate, procedural calculations of a numeric system, symbolic

computing is involved with the interpretation of data to form information, cognitive and qualitative models, and non-deterministic solution processes founded on heuristics and search strategies. The combination of numeric and symbolic programming techniques may be used to approach engineering problems, such as design, which are not purely algorithmic in nature. As outlined by the American Association for Artificial Intelligence workshop on Coupling Symbolic and Numeric Computing Systems in Knowledge-Based Systems and summarized in [36], coupled systems promise to integrate the explanation and problem-solving capabilities of expert systems with the precision of traditional numeric computing.

The Interactive Knowledge-Based System for High-Speed Spindle Design is not an expert system per se, but the system seeks to extend and enhance the capabilities of numeric programs associated with the preliminary design process. The design system provides an intelligent user interface aimed at providing guidance and understanding for the designer as well as acting as a conceptual filter employed to establish functional performance objects, design constraints, and the rapid evaluation of alternatives. A shallow coupling approach of numeric and symbolic programming techniques is not only acceptable, but preferable, in the spindle design application in order to increase program performance and execution speed while limiting hardware requirements.

3.4 AI Applications in Engineering

Most successful applications of expert systems in industry have been applied to problems requiring interpretation, diagnosis, monitoring, instruction, or planning. For the use of this project, only expert systems applied to engineering activities will be reviewed although many of the system structures and methods apply universally. As mentioned earlier, Dym [15], has extensively surveyed expert system approaches applied to Computer-Aided Engineering (CAE). Beyond expert systems, the review characterizes new technology, hardware, and software which have tremendous implications and potential for the sharing of knowledge and the integration of design and manufacturing.

With the potential of increasing productivity and capturing knowledge from an aging workforce, researchers and industry have begun to develop domains in manufacturing and design. Knowledge is increasingly viewed as a corporate resource. Potential applications of expert systems being developed include:

- analysis assistants and intelligent interfaces
- on-line selection of machining parameters
- design of molds and dies
- generative process planning
- assembly planning

- factory management
- cost estimation
- mechanical diagnosis.

The very nature of real engineering, manufacturing and design problems has challenged the implementation of expert systems in the past. Typically these problems are ill-posed and open-ended lacking well-defined objective functions which can be optimized. As Lu suggests in [41], there is a need to integrate "deep" deterministic knowledge with heuristic knowledge obtained from human experience and provide a problem solving mechanism for the integration of various disciplines. Deep systems derive their conclusions from models of phenomena in their domain using basic scientific principles embedded in the models [11]. A deep system may be described as an "Analytical Deterministic" model and differentiated from an "Empirical Stochastic" model which attempts to describe phenomena which is strongly stochastic in nature.[1]

Chryssolouris and Wright summarize knowledge-based systems in manufacturing [6]. The systems fall within the broad categories of design, process planning, scheduling, and process control. The most common application of expert systems in manufacturing is in process planning. STRIPS is a system utilizing a planning-type approach for learning and

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[1] The "Analytical Deterministic" and "Empirical Stochastic" classifications of models were coined by Dr. Jiri Tlusty, University of Florida, March, 1989.

executing generalized robot plans. The system consists of a knowledge base containing IF-THEN rules.

GARI is a general purpose process planner developed at the University of Grenoble in France. The system is written in LISP and uses a forward-chaining inference engine to activate rules based on the interactive addition of assertions into its database.

The AGFPO (Automatic Generation of Forming Only Process Outlines) demonstrates an alternative approach for knowledge-based systems. The program creates deep drawing process plans founded on a generate-and-test procedure. Each potential solution which is generated is then tested against the requirements of the problem. This method may also used in the development of a design system.

An example of a scheduling knowledge-based system for manufacturing is ISIS, which is a job shop scheduler. As with most schedulers, ISIS schedules machining operations based on a search guided by economic constraints, technical considerations, resource limitations, and personal requirements.

With the manufacturing environment continually shifting to smaller batch production with increased product variety, the potential for decision making systems applied to flexible manufacturing has increased significantly. MADEMA (Manufacturing Decision Making) is one such knowledge-based system which can make decisions in a changing manufacturing environment. The system combines artificial intelligence with

operations research methods to develop a decision making process. The system makes use of programmed modularity and exhibits a hierarchical structure representing the manufacturing system with several levels of control.

The preceding examples illustrate only a few of the applications of knowledge-based system in manufacturing but they do indicate general structures utilized and suggest some of the demands of expert systems. Sohlenius and Kjellberg review the potential of AI in manufacturing and summarize some of the demands of future computer aids [63]:

- the system must be based on knowledge representation and provide knowledge interrogation;
- product modeling must be integrated with AI knowledge-engineering techniques;
- the system must be supported by graphical and problem oriented symbols;
- the system must be adaptable to the knowledge of the engineer;
- the computer must be developed and utilized for problem solving, communication, education, and knowledge enhancement;
- the system must aid in the integration of independent product development.

In the past, few attempts have been made to address the need for knowledge-based systems in engineering design although the need has been clearly experienced and often described. Lu explains that problems that fall into the category of formulation require much further investigation, particularly concept development and the early design stages in engineering [41]. Winston [80], concisely expresses this position when he states, "In engineering, computers should check design rules, recall relevant precedent designs, offer suggestions, and otherwise help create new products." Although new commercially available products such as Iconnex's Mechanical Engineering Workbench and Cognition's ConceptStation have begun to address the need for computer-aided concept design, the application of expert system technologies has been limited. Cognition does offer an expert advisor for manufacturability.

For the most part, most knowledge-based systems in design have taken the form of expert advisors for analysis. Many people have developed expert systems for finite element modeling, analysis, and optimization similar to Chen's system developed at the University of Florida [4]. Draisin and Peter [14], describe the development and use of an intelligent interface for design and simulation called Procon (Production Controller). The interface is an example of data-driven programming which aids the user in choosing the proper simulation codes and in the required data preparation for complex nuclear weapons design at the Los Alamos National

Laboratory. The system has been expanded to include the Designer's Apprentice [1]. The systems help users incorporate more factors in the preliminary stages of design and define performance objectives earlier in order to avoid costly attempts to optimize subsystems later in the design process. The system provides a unified interface to a variety of tools and analysis programs.

The structure and methodology of a design system based on an expert system model is described by Kinoglu [34]. The prototype model employs constraint propagation in order to establish the interaction and relations between the various stages of the design process. This design model has been enhanced in order to form the foundation of the Interactive Knowledge-Based System for High-Speed Spindle Design. The use of constraint propagation is further developed by Orelup and Dixon [46], who propose the use of Dominic II, a domain independent design system. Dominic II is based on an iterative redesign methodology which may be limited through the early definition of design objectives and is not intended for design innovation.

Finally, the available and missing tools required for manufacturing and engineering systems are reviewed in detail in Hatvany [27] and Markus and Hatvany [42]. Both outlines describe that one of the worst bottlenecks in engineering today is experienced in the design phase. The surveys express the need to develop better conceptual models through the clear specification and definition of design parameters, design

logic rules, dynamic constraints, and spatial constraints. This is a fundamental motivation for the application description capabilities in the developed Spindle Design System.

As Nooter Corporation experienced with the development of an expert system that designs bolted joints for large pressure vessels and heat exchangers, the proper application of an expert system in a narrow field yields predictable results. Nooter's expert system resulted in the standardization of flange configurations with a reduction of design and production costs. One unexpected benefit experienced by Nooter which often results from the proposed development of an expert system is a better understanding and definition of the problem addressed, product function, or design process [38].

CHAPTER 4

SPINDLE DESIGN

Machine tools are one of the most important means of production for manufacturing industries. In many machining applications relatively large volumes of metal are removed by milling. As a result of the increased demands in both performance and precision, the spindle designer is required to demonstrate an in-depth knowledge of the capabilities and capacities of the components and elements of machine tools. For the efficient operation on any system, the design configuration is of great importance. The designer must capture both the form and function of a product while eliminating the potential of costly modifications and subsequent retrofits.

Because of the increasing demands for higher productivity, accuracy, and reliability of machine tools at a minimum cost, new designs satisfying conflicting functional requirements and design constraints must be explored. Design methodologies are presently illustrating an increased use of computer-aided techniques based on rigorous analysis for the evaluation and verification of design alternatives. Although well established numeric tools have been developed for the

modeling and analysis of structures such as spindles, little concentration on the representation of present design knowledge and the definition of required spindle performance has been demonstrated.

The spindle-bearing system is one of the most significant elements of the machine tool because the spindle often critically influences the stability of the structure and drive systems. The dimensional accuracy and surface finish of the machined workpiece, as well as the metal removal rate of the machine tool, are directly governed by the static, dynamic, and thermal behavior of the spindle-bearing system.

Manufacturers of state-of-the-art, high-speed milling machinery are constantly updating and enhancing spindle designs to provide improved performance. In general, the ideal spindle for high-speed milling applications operates at high rotational speeds while providing sufficient dynamic stiffness to maintain a high power utilization. Current high-speed spindles and attachments do not provide these ideal characteristics. Poor spindle design may therefore result in excessive manufacturing costs due to the rejection of defective or out of tolerance parts which cause a loss of both time and material. In addition, process instabilities may substantially limit the machine tool productivity resulting in a forfeiture of profits for the machine user. The final selection of a spindle must be a compromise based on the machining requirements and the part attributes associated with specific manufacturing applications.

4.1 High-Speed Milling

Due to the rapid development of cutting tool materials including high-speed steel, carbide, ceramic, and diamond, the machine tool industry is experiencing a continued trend to high-speed spindles. Specific terminology has been defined to describe the regions by which the machining speed is referred [19]. The category of "high-speed machining" formally corresponds to peripheral speeds of the cutter from 600-1800 m/min (corresponding approximately to 2000-6000 sfpm). Processes involving cutting speeds from 1800 m/min to 18000 m/min are considered "very-high speed machining," and the limited applications using even higher speeds are designated as "ultra-high speed" or "ballistic machining." In practical high-speed milling applications, the machining performance is considerably more restricted [69]. Spindle speeds of 1800-3600 rpm with standard endmills (10-50 mm in diameter) and facemills (100-250 mm) are common. Spindle powers on commercially available machine tools vary but generally range from 7-30 kW (roughly 10-40 Hp). Metal removal rates up to 165 cm^3/min (10 in^3/min) are typical, and machining times varying significantly from a few minutes up to 24 hours. The introduction of Si_3Ni_4 based tool materials has enabled cutting speeds up to 1520 m/min (5000 sfpm), in the machining of gray cast iron. The maximum cutting speeds of aluminum alloys are generally not limited by tool life or

cutting forces, but the technology of high-speed spindles. The speeds are restricted by the rotational speed capacity of the bearings and the spindle characteristics.

A more general definition of high-speed machining is used in the context of this dissertation. Spindle speeds resulting in the cutter tooth impact frequencies defined by

$$f_t = N\ m\ /\ 60$$

where

f_t = tooth impact frequency (Hz),

N = rotational speed of the spindle (rpm),

m = number of teeth on the cutter,

which approach or exceed the dominant natural frequency of the machine tool/workpiece system are considered "high-speed." Tooth impact frequencies which may significantly influence the structural dynamics of the machine tool typically vary from 400-800 Hz.

The justification for directing efforts into high-speed, high-power machining is the dramatic improvements in the productivity of the metal cutting operation. There are several attributes of high-speed machining which lead to elevated overall productivity. Provided sufficient drive power is available to the spindle, large axial and radial immersions allow increased metal removal rates. The corresponding reduction in machining times per part permits

the distribution of fixed capital costs associated with the machine tool over a greater number of parts, therefore reducing the cost per part (providing more parts are manufactured). In addition, in processes such as the endmilling of aluminum, the cutting efficiency as measured by the power consumed per volume of metal removed increases with the spindle speed.

Although the machining time of a product represents a varied percentage of the total production cycle depending on the manufacturing application, there are a variety of benefits which may be associated with high-speed machining. These potential benefits may result in a reduction in the total production cost per part.

It has been suggested that side loads between the cutter and the workpiece can be significantly lower in high-speed milling [35]. Consequently, a reduction of the deflection of the cutter and deformation of the workpiece during the cutting process results in improved accuracy. A reduction in cutter load within limits additionally leads to decreased tool wear and less frequent re-sharpening. Another advantage of high-speed machining described is the reduced heat flux into both the workpiece and the cutter. Because of the high chip velocity, there is very little time for the conduction of heat into either the tool or workpiece. The majority of heat generated in the cutting process is carried away by the chips. Machining errors associated with significant thermal distortion of the workpiece and the tool are minimized and the

chance of permanent distortion in thin sections or detrimental residual stresses are reduced. As a result, post-machining stress-relieving may be eliminated if distortion and surface hardening is limited.

Additional manufacturing activities may be eliminated by the utilization of high-speed milling. The considerable time and effort expended on the design, development, and production of a casting or forging may be avoided for some components by machining the parts directly from block, sheet, or bar stock. The surface finish produced for a given metal removal rate may be greatly improved through the use of high-speed machining. The time and cost of the grinding, de-burring, and manual finishing of chatter marks due to unstable machining may be eliminated in many applications.

A methodology for determining the machine tool requirements for high-speed machining are discussed in [37]. A set of detailed, object-oriented diagrams and charts are presented to represent the functional model requirements of high-speed machining. The paper summarizes the benefits of high-speed machining as well as presents a questionnaire which may be used to establish the requirements and specification of a high-speed machining system based on the workpiece characterization and economic analysis. A similar methodology is employed by the Spindle Design System in order to direct the design of the spindle based on the performance requirements extracted from machining application descriptions.

4.2 Ideal Spindle-Bearing Characteristics

The main functions of the spindle-bearing system is the guidance of the cutting tool with adequate kinematic accuracy and the absorption of externally applied forces with minimum distortions. For high-speed milling applications, these functions are translated into two basic requirements of the spindle to provide high accuracy and facilitate stable machining over a wide range of rotational speeds. The spindle accuracy is affected by the rotational accuracy provided by the proper selection and installation of precision bearings, and the workpiece accuracy. Because the quality of the workpiece is of primary concern in manufacturing, the spindle-bearing performance should dictate the design process.

The following ideal characteristics of a high-speed spindle-bearing system are summarized by Pruvot [50]:

- the spindle must meet the application performance requirements in terms of speed, capacity, and stiffness;
- the spindle should be stiff when subjected to external loads, but the bearings should have as low as possible stiffness when subjected to internal loads;
- just enough bearing lubricant should be provided for an elasto-hydrodynamic film to build up, in order to minimize power consumption and allow air to cool the spindle components;

- the spindle should be insensitive to housing cooling to avoid bearing seizure;
- the components should be easily manufactured;
- the original adjustments should be easily made and reliable.

These ideal characteristics are designed into the spindle by the proper selection of components and the spindle-bearing configuration. A variety of attributes influence the design or selection of a high-speed spindle. Through the interactive description of machining applications, the workpiece material properties and performance requirements of the spindle are defined. Coupled with the direct specification of geometric constraints, the spindle characteristics are established based on the attributes summarized below:

- spindle power and speed range
- type and number of bearings
- diameter of spindle shaft
- length of overhanging shaft and bearing span
- taper size and type
- spindle mounting orientation
- type and size of cutter
- axial and radial stiffness
- natural frequencies and critical speeds.

4.3 Spindle Bearings

The selection of a particular bearing type and configuration is governed by the specification of the bearing duties in terms of running speed, load-carrying capacity, direction of externally applied forces, life, operating temperatures, mounting considerations, maintenance and geometric constraints. Spindle bearings have a particularly difficult specification to fill because they are required to meet high running accuracies under a wide range of loading conditions and rotational speeds. The emergence of carbide tools, coated carbides, cubic boron nitride, and polycrystalline diamond tools have permitted increased cutting speeds and cutting forces. The trend has necessitated larger diameter spindles, stiffer bearings, and higher spindle speeds. The DN limit for spindles founded on rolling bearings is in the range of 1-2 million and requires a compromise regarding the spindle stiffness. The common DN speed rating is defined by, $DN=d \cdot N$, where d is the bore diameter of the largest bearing (mm). Hydrostatic and hydrodynamic bearings permit similar high speeds in spindle applications but they generate considerable power loses and thermal problems, [69]. In addition, aerostatic and magnetic systems have been explored but have mainly not been successful due to lower stiffness and high susceptibility to surface contact. The bearing requirements of high speed and stiffness are somewhat

restricted to the machine tool industry. In the majority of applications, such as in jet engines, bearings are installed with clearances or minimal preloads in conjunction with compliant races and housings. The problems of bearing failures and poor performance play a key role in the design of high-speed milling spindles.

The criteria by which spindle-bearing performance is evaluated includes a variety of often conflicting characteristics. As mentioned, the spindle bearings must operate over a wide range of rotational speeds while providing a high degree of accuracy. The portion of the total spindle drive power dissipated in the bearings must be minimal and a satisfactory performance must be maintained over an economically viable lifespan. In order for the bearing requirements to be satisfied, the bearing's operation must generally satisfy the following conditions:

- the combined effect of internal and external loads must be within the bearing load carrying capacity;
- the lubricant films separating the rolling elements from the bearing races must be of sufficient thickness to prevent metal-to-metal contact;
- the temperature rise in the bearings during operation must not result in degradation of the lubricant or failure due to thermal instability.

The basic bearing arrangements and descriptions are reviewed in detail in Deutchman et al. [12] and Weck [79],

with bearing type and size comparisons summarized by Devorak [13]. Fundamental spindle design concepts are based on paired configurations of tapered roller, cylindrical roller, and angular contact ball bearings. Figure 4.1 illustrates the pair layouts of rolling bearings. Although angular contact ball bearings are depicted, any bearings with a non-zero contact angle may employ the layouts.

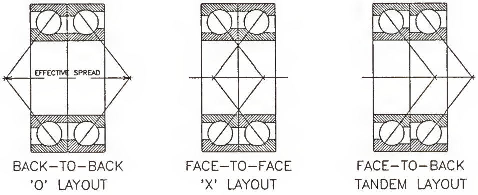


Figure 4.1 Duplex Mounting Arrangements

The back-to-back arrangement shown on the left, illustrates an 'O' layout defined by the contact axes between the rolling elements and races. The intersections of the contact axes with the bearing's centerline delineate the effective spread of the bearing pair. The 'O' layout is characterized by an effective spread greater than the actual width of the duplex arrangement, therefore the configuration provides a good end restraint for the spindle due to the favorable line of action for the force transmission of the balls. In addition the configuration is well suited for the absorption of cantilever loads. The 'X' layout in the center of the figure depicts the

face-to-face bearing pattern which reduces the effective spread of the arrangement to less than the overall width of the duplex. Similar to the 'O' layout, the 'X' layout can effectively support axial loads in both directions, but it is seldom employed in present spindle design. When angular contact ball bearings or tapered roller bearings are mounted in pairs with opposite direction of the axes of the rolling elements, the radial load is nearly evenly distributed between the two bearings. Such double bearings are supplied by the manufacturer in matched pairs for particular applications. Finally, the tandem layout shown on the right, depicts a face-to-back arrangement of a bearing pair. The tandem layout is capable of supporting increased axial loads in one direction, in this case an applied load from the right. The tandem pair is commonly used in spindle applications to form half of an overall 'O' configuration. In conjunction with an opposing tandem pair, the configuration offers effective support of both cantilever and axial loads from both directions. Spindle bearings of high-speed machining centers also modify the configurations to form a forward bearing group which consists of a pair of precision angular contact ball bearings arranged in tandem coupled with a third matching bearing mounted in an 'O' layout relative to the duplex. The spindle-bearing configurations used in the design system will be explained in more detail later in this chapter.

The spindle bearings must maintain two aspects of accuracy during operation. Although a common description of

the accuracy of a bearing rotating under no-load conditions is called the radial run-out, a review of spindle error characterization indicated more specifically that surfaces, not bearings or spindles, exhibit run-out with respect to an axis. Spindle accuracy characteristics are addressed by ANSI standard B89.3.4 for Axes of Rotation. This clarification indicates the importance of the bearing race faces as well as the seatings for the bearings within the spindle housing. For high precision rolling bearings properly mounted in well manufactured spindles, rotational errors of less than 0.1 micrometers are readily available. In order to maintain workpiece accuracy, machining errors during milling must be minimized and the stiffness of the spindle bearings must be maximized. The loads exerted on a bearing are transmitted through the contacts between the rolling elements and the raceways. The geometry of the rolling elements and their material properties indicated how well the bearing can transmit the loads during operation. The contact geometry influences the bearing stiffness but also governs heat generation and power losses. The force-deflection characteristics of a spindle are improved through the preloading of the bearings which distributes externally applied loads over a larger number of rolling elements.

Preload refers to the use on an internal or mounting load on the bearing which eliminates diametral clearance in order to increase the load-deflection behavior. When the initial load is applied, all elements are forced in contact with both

races therefore allowing every rolling element to participate in the support of externally applied forces. The load distribution decreases the transmitted force on each element resulting in a reduced deflection. Radial bearings are preloaded by producing a diametral interference caused by driving the bearing onto a tapered spindle shaft. In the case of angular contact bearings, one of the races is displaced axially, loading the elements with a force proportional to the relative displacement of the races and the contact angle. Different spindle designs, in addition to the affect of preload on bearing stiffness, are described in Hernandez and Chen [29] and Tlusty et al. [72], and may allow the bearing preloads to be variable or fixed. A preload obtained by the adjustment of a nut in one case, maintains a constant distance between opposing bearings and supplies an equal, axial preload. Since the constant distance preload is installed during installation there is no mechanism to maintain the preload. Thermal expansion of the spindle shaft relieves the preload on the bearings arranged in an 'O' configuration, diminishing the load-deflection characteristics and often resulting in instabilities. An alternative preload mechanism maintains a constant force acting on each bearing by means of a spring, hydraulic cylinder, or hydraulic piston which controls the translation of the rear bearing. Although a variety of constant preload designs are available, jamming of the rear bearing set which is assumed to be free to float axially, frequently prevents their proper operation. Fixed

distance preload is mostly employed along with angular contact ball bearings with low contact angles (15°-25°). The low contact angles providing reduced axial stiffness compared to tapered roller bearings are rather tolerant of thermal axial expansions of the spindle.

When considering the application of a preload to improve the load-carrying capacity of bearings, an optimum value which balances the increase in stiffness against the increased temperature caused by rolling friction and loss of bearing life must be determined. A comprehensive discussion of spindle preload behavior is presented in [50]. A spindle model based on the consideration of the heat generated in the bearings due to friction, and the manner in which the heat causes the uneven expansion of the bearing components and spindle housing is presented. In general, it is suggested that the expansion of the bearing inner race and rolling elements exceeds that of the outer race and housing causing thermal instability and increased Hertzian stresses which may result in catastrophic failure of the spindle-bearing system.

Finally, in the high-speed application of bearings for machining, a minimum axial preload must be maintained to prevent gyroscopic motions which may cause the sliding of the rolling element relative to the races. In bearings where the angle α between the rotating plane of the bearing and the direction of contact load axes of the elements are not equal, such as angular contact and thrust bearings, gyroscopic couples are caused by the directional change of the axis of

rotation of rolling elements. As illustrated by the extreme case of a grooved ball thrust bearing in Figure 4.2, a moment is required to change the direction of the ball spinning axis.

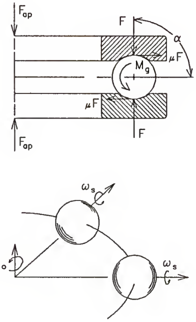


Figure 4.2 Gyroscopic Couple

This required moment is provided by the frictional torque on the ball/race contact. While the ball spins on its axis with an angular velocity of ω_s, the ball also orbits in the rotating plane at ω_o. The gyroscopic moment is defined by M_g, where $M_g=J_b\omega_o\omega_s$, and J_b is the moment of inertia of the ball. If the frictional torque does not provide a sufficient moment to balance the gyroscopic moment, the balls will rotate about the axis of M_g, and produce excessive sliding. Chen [5], suggests that ball sliding may be a primary cause of high-speed bearing failure and once occurring, the thermal

contribution due to sliding may be 20 times that generated due to the ball spinning friction. A speed-dependent, minimum axial preload, corresponding to F_{ap}, which takes into account both the effects of the gyroscopic moment and the centrifugal forces acting on each rolling element tending to force the bearing rings apart, must be determined and maintained.

The conditions stated as necessary for satisfactory bearing operation could all be considered in terms of their impact on the machine tool performance requirements. It is desirable to minimize the power dissipated in the spindle bearings for reasons beyond maximizing the net power available at the spindle for high-speed machining. The power is dissipated in the form of heat which has a detrimental effect on lubricant viscosity and bearing fatigue life. In addition, thermal distortion of the spindle and machine tool structure introduces workpiece inaccuracies. The lifespan of a rolling element bearing is described by its L_{10} fatigue life. The life rating expresses the number of bearing revolutions to failure which 90% of identical bearings will endure. To prevent metal-to-metal contact within the bearing, the film thickness necessary is dependent on the surface finish, lubrication viscosity, and loading of the bearing. Not only the ball/race contact must be properly lubricated but also the interfaces between moving elements such as the ball/cage and race/separator contacts.

The Spindle Design System recommends either grease or oil mist lubrication for high-speed spindles, because forced oil

lubrication results in increased power losses and added complexity to the lubrication system. Grease lubrication is the simplest method of lubricating the spindle bearings. Grease is used extensively because it permits simplified spindle design, less maintenance, and effective protection from contaminants if properly enclosed. Greases used for most spindle operating conditions consist of thickened petroleum and silicon oils. The properties of the greases vary significantly depending on the type, grade, and consistency [13]. The consistency of the lubricating grease is particularly important to avoid excessive churning if the grease is too soft and separation if it is too hard. The sensitivity of the grease to high temperatures (upper limits vary from 149°C-288°C or 300°F-550°F), and the prevention of air cooling, limits the high-speed applications to DN values ranging from 140,000-495,000 depending on seal types [59]. An oil mist lubrication system deposits a stream of atomized oil to the bearing. The oil is generally directed by a small nozzle (0.8-1.3 mm in diameter), against the inner race of the bearing which is difficult to lubricate due to the effect of centrifugal force on the oil. The lubrication system is relatively simple because of low oil flow rates and the fact that oil is passed through the bearing only once and discarded. Compressed air at pressures of 100-200 N/m^2 is dried to eliminate moisture and used to propel the oil. The oil mist lubrication method provides good cooling and supplies sufficient elasto-hydrodynamic films to prevent metal-to-metal contacts.

In the design of high-speed spindles it is important to understand the source of bearing failures and the need for better guidance in the selection of permissible spindle speeds and lubrication methods. The design methodology exhibited in the Spindle Design System makes use of empirical data, DN values, and heuristics as well as analytical models. The design system selects and models bearings to estimate load-deflection characteristics in a detailed manner as discussed in Section 5.6. Quick estimates of the bearing stiffnesses are determined based on the axial or radial preloads, δ_{ap} or δ_{rp}, and the outside and bore diameters of the bearing, D and d, by employing heuristic equations summarized in Table 4.1 [22].

Table 4.1, Bearing Stiffness Equations

Bearing Type	Series	Stiffness Estimate (N/mm)
Thrust Ball Bearing	511,512	$593{,}000(D+d)\sqrt{\|\delta_{ap}\|}/\sqrt{(D-d)}$
Double Direction AC Thrust Ball Bearing	234(00)	$532{,}000(D+d)\sqrt{\|\delta_{ap}\|}/\sqrt{(D-d)}$
Double Row Cylindr. Roller Bearing	NN30 NNU49	$35{,}400\|\delta_{rp}\|^{1/9}d^{10/9}$ $48{,}700\|\delta_{rp}\|^{1/9}d^{10/9}$
Tapered Roller Bearing	30211	$42{,}100\|\delta_{rp}\|^{1/9}d^{10/9}$
Angular Contact Ball Bearing	719C 719AC 70C 70AC 72C 72AC	65,900 55,900 55,200 47,100 55,200 47,100 $\cdot \dfrac{(D+d)\sqrt{\delta_{ap}}}{(D-d)^{3/8}}$

A deterministic evaluation of limiting factors influencing both the thermal and dynamic characteristics of bearings is under develop. The previously referenced work at the University of Florida is such an example [5].

4.4 High-Speed Milling Dynamics

The theoretical and experimental analysis of high-speed milling dynamics has reached an advanced state in which the limit of stability and the accuracy of machining may be deterministically predicted. The self-excited vibrations which occur during the milling of metals is generally referred to as "chatter." The full power utilization of high-speed spindles is frequently limited by the onset of chatter. As a result, unacceptably large vibrations may develop between the tool and the workpiece potentially damaging the workpiece, cutting tool, and even the spindle. Because of the limitation on productivity and post-machining costs attributed to chatter, considerable research and development efforts have been devoted to the design and analysis of high-speed spindles. A deeper understanding of the dynamic mechanisms causing chatter and the special aspects which lead to improved stability in milling must be exhibited by the spindle designer.

Tlusty [69,70] presents a comprehensive description of the theory of chatter, as well as several methods to increase the stability of machining against chatter. For spindle-

bearing systems where the relative transfer function between the tool and the workpiece demonstrates one dominant mode (resonant frequency), well separated from other modes, the system may be modeled in its simplest form by a single degree of freedom model described in Section 5.3. From the milling process model and the specification of machining parameters, the performance requirements of a high-speed spindle may be established.

A feature common to most milling machines is that a cutting tool mounted in overhang within a spindle represents the most flexible part of the machine tool structure. This flexibility is characterized by the relative transfer function measured between the tool and workpiece and is decisive for the determination of chatter. The principle mechanism of the process limiting chatter is self-excited vibration resulting from a regeneration of waviness of the workpiece surface. As the present cutting tooth encounters the undulations left behind by the preceding teeth, displacement of the cutter leaves another wavy surface behind. The relative amplitudes of the consequent vibrations depend on the phasing between the undulations of the present and preceding surfaces. An unfavorable phasing will result in increased vibratory response causing process instability in the form of chatter. On the other hand, favorable phasing will result in diminished vibration and a stable milling process.

The main features distinctive for chatter in milling are discussed by Villers [76], along with a simulation program

used to describe the milling process. Dynamic characteristics of the milling tool and spindle are presented with practical consequences for both the design of high-speed spindles and the selection of cutting parameters in machining. Stability lobe phenomena and an explanation for the generation of damping during the cutting process (ie. process damping), are introduced.

Smith [62], explains computed and simulated stability lobes and presents an algorithm for automatically selecting the optimum spindle speed to produce elevated levels of stability allowing increased metal removal rates. The relation between the transfer function and the stability lobe are represented in Figure 4.3. The axial depth of cut at the limit of stability (b_{lim}), may be represented by

$$b_{lim} = \frac{-1}{2\ K_s\ Re[G]}$$

where K_s is the cutting stiffness (specific power) of the workpiece material and Re[G] is the real part of the oriented transfer function of the system illustrated in Figure 4.3 a). The mapping of the limiting axial depth of cut from the negative part of Re[G] to the corresponding spindle speed generates a division described as the stability lobe. The stability lobe defines the unstable region of axial depths of cut at varying spindle speeds above the lobe from the stable depths below. Several of the lobes which correspond to the

A)

B)

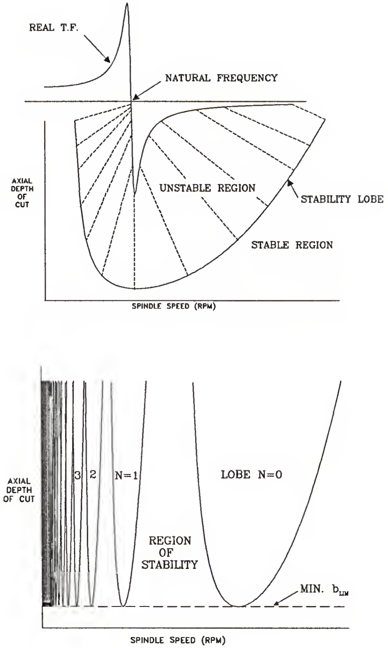


Figure 4.3 Stability Lobe Generation

various number of complete waves (N), between subsequent teeth on the cut surface are computed to form the complete stability plot in Figure 4.3 b). Smith proposes a method to direct the spindle speed to the optimum range which corresponds to the

region of stability. The algorithm selects a spindle speed just below the natural frequency of the system coinciding to the left most point on the lobe. At this speed, the passage of the cutter teeth are in phase with the chatter vibration therefore maintaining a constant chip thickness and cutting force. Although the system is driven at frequencies near the natural frequency, resonant forced vibration is inhibited due to disruption of the vibration development caused by the regeneration of waviness. The procedure offers a valuable method to stabilize the milling process and increase productivity without modifying the original cutting paths generated.

An additional source of machining stability is process damping. The presence of process damping determines the ability of the cutter to regenerate surface waviness. In the case of low cutting speeds and high vibration frequencies, high damping is exhibited therefore allowing stable machining at greater axial depths than at higher cutting speeds. Because of the short surface wavelength, interference between the flank of the tooth and the slope of the undulated surface produces a normal damping force on the tooth. The mechanism for process damping is illustrated in Figure 4.4. The back rake angle of the tooth is represented by α and the relief angle is depicted by τ. The tool exhibits no vertical velocity at locations A,C, and E. As the tool moves down the surface from A to B, the increased normal cutting thrust force (ΔF) caused by the additional interference between the surface

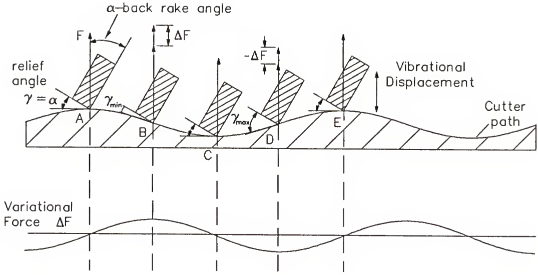


Figure 4.4 Process Damping

and the tool acts against the downward velocity of motion and generates damping. The converse situation occurs as the tool reaches its maximum upward velocity at location D while the maximum relief angle reduces the normal force (F). The corresponding variation of the normal force is shown below the cutting path. The 90° phase lag of the variational force to the motion of the cutter illustrates its damping force behavior.

The surface wavelength defined by the following equation may be utilized as an indicator of process damping,

$$\lambda = v/f_c$$

where

λ = the surface wavelength

v = the peripheral surface velocity of the cutter

f_c = the chatter frequency (Hz).

Further investigation establishing a good correspondence between the spindle speed and process damping may supplement the use of stability lobes in order to stabilize machining and influence the future design of high-speed spindles.

4.5 Spindle Behavior and Optimization

As previously explained, the dimensional accuracy and surface finish of the workpiece as well as the metal removal rate of the machine tool are directly governed by the static, dynamic, and thermal behavior of the spindle-bearing system. The design process involves the choice of an appropriate spindle-bearing configuration and the geometric optimization of design variables providing a minimum flexibility at the tool/workpiece interface. The static behavior of a machine tool is defined by the total deflection of the cutting tool at the point of force application. There are a number of contributory elements of the machine tool which influence the static stiffness measured at the tool, although practically speaking, the spindle is frequently the critical source of flexibilities. These flexibilities may be caused by spindle and bearing flexure, deflections of the spindle housing, and flexure of the tool holder and cutter. Assuming a robust housing, the maximum static stiffness is achieved by the selection of stiff bearings and the optimum overhang to bearing span ratio (A/B). In the static case, the largest

bearing bore diameters which will satisfy the load and speed requirements are selected, and the radial stiffness of the front and rear bearing groups are estimated. The optimal bearing spacing is determined through a number of methods.

A functional optimization approach by Singhvi et al. [60], is aimed at minimizing total cross compliances and mean squared responses between the cutting tool and the workpiece. The required stiffness to achieve the desired threshold of stability is considered. The static optimization is based on the A/B ratio, the flexibilities of the front and rear bearings, $1/K_f$ and $1/K_r$, and the stiffness ratio K_f/K_r. The functionally optimal design is evolved through the use of a series of design optimization charts and a procedure which aids the designer in developing an understanding of influence relationships between the variables. It is suggested that the design parameters can normally be selected in such a combination that the design constraints are satisfied with A/B ratio ranging from 0.3-4.0. In addition, the paper discusses the total and effective stiffness of various bearing types and presents a table of simplified equations.

A study which evaluates the merits of various spindle bearing arrangements for achieving the best compromise to meet the demands of speed, power, and accuracy is presented in [22]. The study indicates that as a good approximation, the support of a spindle can generally be considered as a two bearing arrangement. The bearing stiffnesses are estimated for zero-clearance, preloaded spindle bearings based on the

preloads and diameters. An optimum bearing span for minimum radial deflection assuming negligible shear stress is given by

$$B = [6 \; E \; J \; (1/K_f + 1/K_r)]^{1/3}$$

where E is the modulus of elasticity and J represents the spindle average moment of inertia.

Finally, a variety of nomograms have been developed to determine the statically optimum bearing spacing and the optimum spindle dimensions of solid, uniform shafts, [17][48][79]. Most static analyses of the spindle describe the total deflection of the tool in the cutting zone (δ_T), by the deflection components attributed to the bearings (δ_B), and the spindle shaft (δ_S) as illustrated in Figure 4.5. The geometry of the spindle is represented in a simplified manner by the front, inner, and rear shaft diameters as shown.

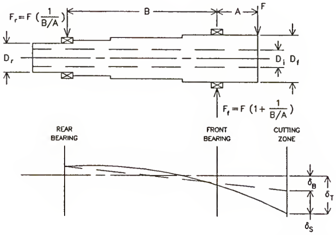


Figure 4.5 Spindle Deflection Components

The general static behavior of the spindle-bearing system assuming a constant applied force (F), may be illustrated by Figure 4.6. The figure demonstrates the relationships between the bearing deflection, spindle deflection, and the total deflection measured at the tool as a function of the varying bearing span. The figure more accurately describes the spindle behavior with a facemill as opposed to a relatively flexible endmill. The spindle component of the total deflection varies linearly with the bearing span, where the bearing component influence the deflection as B^2. Independent of the method, the static optimization of the design variables should define a bearing span in the general design region depicted and minimize the overall spindle flexibility.

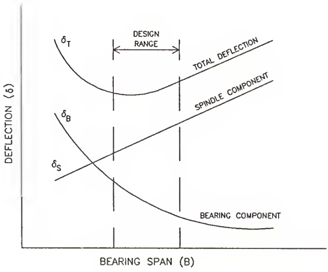


Figure 4.6 Optimum Bearing Spacing

The dynamic characteristics of the spindle-bearing system have considerable influence upon the geometric accuracy and

surface finish of the workpiece as well as on the chatter-free cutting power available. The dynamic behavior of the spindle is represented by the transfer function measured at the cutting tool. The criteria for the dynamic behavior are the resonance frequencies, resonance amplitudes, mode shapes, and the system damping. The dynamic stiffness of the spindle is of considerable significance in avoiding regenerative chatter which usually occurs close to one of the resonance frequencies of the spindle. The frequency behavior of the spindle is dependant on the mass and stiffness distribution of the system, $f(K,M)$. The vibration amplitude which directly affects the limit of stability in machining, is determined by the stiffness and damping properties of the spindle-bearing system, $f(K,\zeta)$.

Numerous methods have been proposed to improve the dynamic characteristics of high-speed spindles. The design parameters introduced have included the mass and stiffness distribution (defined by diameters, density, and second moment of inertia), the modulus of elasticity of the shaft material, the bearing stiffness and configuration, amplitude and position of the cutting and driving forces, the spindle length and overhang, and damping. The optimization of the spindle dynamics should consider the complete system dynamics and the excitation environments. It has been suggested that for multipurpose machines, dominant natural frequencies and excitation frequencies should be separated considering weighted fractional utilization. Other optimization criterion

include providing the highest possible natural frequencies, or minimizing mean squared responses under the assembly of power spectrum or excitation environments [60]. Multiple objective functions may also be employed as in Reddy and Sharan [53], where a method for designing lathe spindles based on minimizing the workpiece response is described. A matrix reduction technique is used for the static and dynamic analysis of the system. The paper suggests that the design of a machine tool spindle is a complex problem and that performance requirements are strongly dependent on the actual machining applications.

Design recommendations for the improvement of the dynamic behavior of the spindle-bearing system are rather general and primarily based on experimentally established trends. For example, Weck [79], simply suggests improvement of dynamic behavior by retaining the spindle front bearing position by the use of several preloaded bearings and the damping of bending vibrations with a damping element.

The dynamic behavior may be marginally improved by the methods mentioned because the traditional modifications of stiffness and mass may be practically varied over a restricted range. Although numeric results are limited, the most significant improvement in the dynamics of spindles lies in the least understood area of damping. Additional damping can be introduced through internal and external means. Internal damping has been suggested by the use of modified sandwich structures within the spindle, varied bearing preload, and

the use of a third bearing of limited stiffness but higher damping, placed between the front and rear bearing supports. From a number of experiments it is indicated that the bearing damping is mainly the result of velocity proportional, viscous damping, [79]. The internal damping means have produced marginal results and are therefore prohibitive relative to the increased complexity of the spindle design and manufacturing. External means of damping have been more successful and have concentrated on the application of tuned damper systems attached to the spindle at a location exhibiting larger deflections. Present research in the Machine Tool Laboratory at the University of Florida by J. Tlusty and W. Cobb have demonstrated considerable dynamic improvement for spindle attachments and boring bars through the addition of tuned dampers. Practical applications for spindle extensions used in die machining have resulted in metal removal rates increased 3-5 times. A pretuned damping ring mounted within the spindle extension housing is used to oppose the in phase motion of the housing and the spindle shaft. Optimization of the mass ratio of the damper and reflected modal mass, the damping characteristics (ζ), and the modal frequencies have indicated the feasibility of dynamic improvements of greater than 10 times. These improvements are implemented by the balancing of the negative real transfer function of the dominant spindle mode and the applied damper. Results have indicated that a slightly overtuned damper can be added to reduce the critically limiting spindle mode.

Finally, the thermal behavior of the spindle is also of particular interest because as Pruvot states [50], often the greatest part of machine down time is due to the failure of the spindle bearings resulting from thermal instability of the bearings, spindle, and housing. Pruvot suggests the use of bearing rolling elements which are as small as possible and made of materials with a low coefficient of linear expansion such as silicon nitride or ceramics. The thermal model in [50] indicates that in general, increased spindle stiffness also implies a thermally less stable system. As indicated by the negative feedback in the thermal model, the increase of the spindle housing temperature has a stabilizing effect on the system as a result of the corresponding reduction of bearing preload, therefore no attempt should be made to cool the housing.

The primary source of heat generation in spindles is the bearings. The bearing temperatures increase as a strong function of the spindle speed. The increase bearing temperature affects the spindle performance in mainly two ways. First, the temperature rise reduces the viscosity of the oil in the film between the rolling element and the raceways. The film thickness diminishes as a result of the loss of oil viscosity and the load carrying capacity of the bearing is diminished. If the lubricating film is eliminated due to excessive temperatures, metal-to-metal contact will result causing the eventual failure of the bearing. The second manner in which the heat generation affects the spindle

performance is due to the geometry and varying thermal capacities of the spindle components. As the principle source of heat generation, the bearing temperature increases more rapidly than the spindle shaft or the surrounding housing. Uneven thermal expansions produce a diametral interference in the bearings and result in inducted bearing preloads. The increased internal bearing loads precipitate elevated stresses which may reduce the life of the bearing. The poor thermal behavior represented by the thermal distortions of the spindle components may also result in a loss of machining accuracy. It should be noted that due to the small heat capacity of the rolling elements most of induced bearing loads are transient, therefore the steady state thermal behavior is dictated by the direction of the heat flow.

The experimental investigations of two spindle configurations are presented by Hernandez [28]. One spindle configuration was based on double row cylindrical roller bearings, while the other design was founded on taper roller bearings with a constant preload mechanism. The spindle performance was evaluated in terms of operating temperatures, cooling requirements, and power losses during high-speed operation. Special emphasis was given to the forced oil lubrication method and the search for an optimum oil flow rate. Empirical equations are formulated to describe the behavior of the tested spindles. The concluding design recommendations of the study will be included later in this chapter.

4.6 Design Configurations

For many machine tools the vibrational characteristics are largely dictated by the configuration of the spindle-bearing system. A review of spindle designs and analysis has indicated that basically five or six general configurations are primarily used. Most proposed spindle designs based on rolling elements seem to be a modification of one of the configurations depicted below in Figure 4.7.

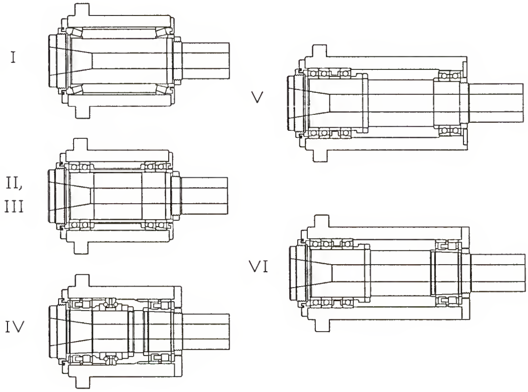


Figure 4.7 Spindle Configurations

For this reason, the Spindle Design System employs these configurations as the conceptual basis for the spindle design process and utilizes Waleckx's study [78] to determine the most appropriate configuration which will achieve the best compromise to the spindle performance requirements.

Configuration I is a simple design incorporating taper roller bearings mounted in an overall 'O' layout. The design requires delicate internal clearance adjustment because the taper roller bearings are more sensitive to preload. Because of the kinematics of the design, the configuration is susceptible to heat generation and thermal expansion. For this reason the design is primarily used for very short spindles with good lubrication. The speed limit is conventionally around DN=200,000. The advantage of the taper roller bearing configuration is that a thermosymmetric design may significantly reduce the thermal effects of the spindle. Figure 4.8 illustrates the thermosymmetric design, characterized by the axes of the rolling elements intersecting at a common point on the spindle axis [79].

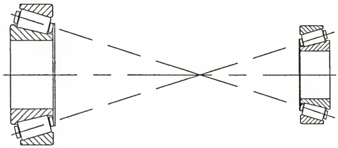


Figure 4.8 Thermosymmetric Spindle Design

The design reduces the thermally activated preload changes of the taper roller bearings because an equal change in radial and axial temperatures will cause any variation in the axial preload to be cancelled out by a compensating change in the radial preload.

Configuration II and III are also a relatively simple designs and in themselves represent only a slight variation. Design II utilizes two matched sets of single row angular contact ball bearings with a contact angle of 25°, (often SKF series 70). The configuration provides high running accuracy due to the use of the matched sets. The bearing are arranged back-to-back providing an overall 'O' layout. The bearing sets are separated by spacer sleeves of "exactly" equal length so that adjustments of the bearings made during manufacturing are not disturbed. The design provides excellent axial support and is the primary alternative to configuration I for very short and short spindles. The speed limit is determined by the lubrication method and seal type, but DN values in excess of 480,000 are readily obtainable.

Configuration III is identical to II with the exception of the front bearing contact angle. The work end of the spindle utilizes a 15° angular contact ball bearing while the drive end maintains the 25° bearing. The spindle is used for short spindle applications where predominantly radial loads are incurred. The lower contact angle in the front provides increase radial stiffness while the rear incorporating the larger contact angle accommodates axial forces induced at the

loaded work end. The design may not be used in very short spindle applications because the different contact angles of the opposing bearings results in different ball and cage speeds which may excessively work the lubricant and accelerate its migration.

The well established and proven spindle configuration represented by design IV was developed by SKF about 1955. The design separated the radial and axial support functions of the spindle. The radial bearings are mounted with interference fits and the radial preload is determined by the axial location of the inner rings on the tapered shaft seatings. The double row cylindrical roller bearings (SKF series NN30K), at the front and rear of the spindle provide excellent radial stiffness with good kinematics and low friction moment. The axial support of the spindle is provided by means of a double direction angular contact thrust ball bearing (SKF 2344-00) or 60° Radiax bearing (234420 BMI). The outer diameter of the thrust bearing requires the same bore seating diameter in the housing as the adjacent front bearing. The thrust bearing is preloaded by using an internal nut on the spindle shaft which axially presses both rows of balls together. The preload is applied during the assembly of the spindle and cannot be modified intentionally without disassembly. The arrangement provides high stiffness characteristics and may be implemented in short to long, slim spindle applications. The maximum rotational speed of the spindle configuration is primarily limited by the thrust bearing because of thermal expansions

and centrifugal forces eliminating the bearing preload. The configuration is frequently used up to DN=475,000 but has been successfully employed up to 900,000 with the use of chilled, forced oil lubrication [28].

Configurations V and VI are frequently employed designs which are functionally similar. Three matched single row angular contact ball bearings with a contact angle of 15° (SKF series 70C), are used at the work end of design V. The front bearing arrangement forms an 'O' layout and supports cantilever loads well due to the increased effective bearing spread. Two similar bearings are paired back-to-back at the drive end of the spindle. The design is primarily used in larger spindle applications therefore the housing seatings are stepped in relation to each other enabling machining from one direction and facilitating accurate alignment. The load is distributed nearly evenly between the bearings of a set and is capable of high rotational speeds with good stiffness despite the exclusive use of low angle ball bearings. The rear bearings are non-locating and are mounted with axial freedom. The spindle has much better radial than axial load capacity as a result.

Configuration VI utilizes 25° angular contact ball bearings at the work end of the spindle in the same arrangement as design V. The rear bearing set consists of a double row cylindrical roller bearing which are also allowed to float axially within the housing. The use of a large contact angle in the front bearing set provides better axial

support while in combination with the rear roller bearings maintains a high radial stiffness. Published operating speed ranges for configurations V and VI are as high as DN=840,000 with oil mist lubrication [59].

4.7 Design Considerations and Guiding Trends

A review of spindle designs, experimental test results, and analysis has indicated the need to develop a design methodology which captures available knowledge and couples the design task with numeric analysis. Due to the nature of design and component information, which is usually related and limited to specific test arrangements, only general design considerations and guiding trends are summarized. The nature of the information in this section indicates the need for better guidance in choosing spindle components and speeds as well as the selection of adequate lubrication methods instead of relying on empirical data and DN values.

The practical range of diameters for a machine tool spindle is rather narrow with the size being dependant on the design and performance requirements. Design considerations include the speed range, tool and workpiece diameter range, drive system, spindle head size and geometric constraints. The performance requirements are influenced by the selection of bearing types and arrangements, parameters such as stiffness, load capacity, power loss, operating temperatures, mounting considerations, and maintenance costs. The overall

spindle design must be evaluated by its ability to satisfy the machining applications. This goal is generally translated into providing adequate accuracy and sufficient stiffness to maintain stable machining. The stiffness of the spindle-bearing configuration is of primary concern and is maximized by choosing the correct bearing types, preloads, and spacing, as well as the spindle shaft and housing specifications.

In the design of high-speed spindles for milling, the factors which determine the speed at which the spindle can operate is the increase in temperature, method of lubrication, accuracy of adjacent components, bearing preload in operation, external loads, shape of the housing, attachment of tools, and drive location. As indicated by experiments on standard bearing arrangements with grease lubrication, [3], spindles may be viably operated at speeds in excess of catalogue limiting values. Only grease-lubricated spindles were tested because the greases are better defined than lubricating oils. The study indicates that the accuracy of the components adjacent to the bearings significantly influences the temperature rise and therefore the achievable speeds. For high-speed operation the components of the spindle must be machined with specified accuracy. The Spindle Design System concentrates on the configuration and evaluation of design concepts and is not involved with the design detailing. Form tolerances for the detailed design of the spindle and housing are depicted in Figure 4.9 [3]. The form tolerances, symbols, and reference surface should be manufactured in accordance with ISO/R 1101 and conicity determined by ISO 3040.

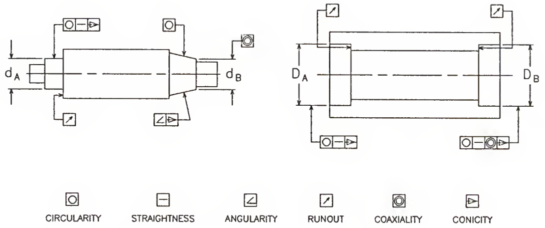


Figure 4.9 Detailed Design for Spindle and Housing

In addition to the detailed design of the spindle and housing, the type of seal design must be determined. Rubbing seals with grease lubrication may be used up to DN=140,000. At higher rotational speeds, a non-contact seal must be employed due to the friction torque and heat generation. The labyrinth seal may be used with grease lubrication up to DN=450,000 and then with oil mist lubrication for high speeds [59].

The remainder of this section will summarize influential design trends in a general manner. During cutting, the spindle undergoes a complex loading which may be resolved into axial (thrust), tangential (twist), and radial components. The radial component is the most significant because the deflection due to bending is greatest and has a direct effect on both the dimensional accuracy and surface finish of the workpiece. The deflection in the cutting zone results from

spindle flexure, which increases linearly with the bearing span to overhang ratio (B/A), and bearing compliance, which decreases nonlinearly (B^2), as B increases. Given the spindle-bearing configuration, the bearing set locations may be optimized in order to minimize the flexibility in the cutting zone. If the actual bearing spacing is appreciably greater than the optimal value, the design may be improved by increasing the stiffness of the spindle shaft. If the bearing spacing is less that the optimum value, increasing the stiffness of the bearings will improve the design. Varying cutting forces may change the optimal bearing spacing slightly for roller bearings (5%), but may significantly influence the optimal span for ball bearings (40%). The use of a tandem bearing arrangement at the spindle nose gives an effective center closer to the cutting zone and therefore reduces the static and dynamic flexibilities attributed to the overhang. Some benefit may be gained by the shaft material choices. Cast iron exhibits a damping ratio several times that of mild steel, and the use of composites may provide large increases in damping. Perhaps the most promising design alternative for boring bars, spindles, and attachments is the addition of an auxiliary spring/mass/damper assembly acting as an externally tuned damper system. Potential gains in metal removal rate for spindles exhibiting predominantly one critical mode are an order of magnitude. An obtainable damping range of $0.10 \leq \zeta \leq 0.30$ for the tuned damper has been demonstrated. Additional thermal aspects of the spindle must be also be

considered. Cooling oil passed through the spindle can result in temperature differences and thermal instability during operation. Temperature differences and varied rates of thermal expansion may result in either the loss of bearing preload, or the seizure of the bearings. The use of coolant applied at the spindle nose may necessitate different adjustments for the front and rear bearings.

The choice of the proper bearing types are crucial to the spindle design. The primary demands placed on the bearing arrangement are that it should provide adequate running accuracy and good stiffness characteristics. Several sources act as a good review of trends in the choice of bearing types [13], and characterization of bearing behavior [39,69]. Roller bearings are consistently stiffer than comparable ball bearings. The roller stiffness may be ten times that of the same size ball due to the longer contact length. It has been demonstrated that the stiffness of the roller bearing is influenced primarily by the contact length and that the roller diameter affects the stiffness minimally. The load capabilities of the roller bearings are correspondingly higher than ball bearing but the increased contact length of the roller limits the high-speed applications of the bearing due to greater heat generation. Non-locating roller bearings have high radial stiffness and load capacity in addition to high-speed capabilities. In angular contact ball bearings, the axial and radial bearing stiffness may be influenced by the selection of the contact angle, ball size, or conformity

(decreasing the race curvature). The contact angle has the largest influence on the stiffness of the alternatives above and is the easiest to adjust.

The amount and type of preload may dramatically modify the bearing behavior. Increasing preload in the bearings will provide improved stiffness and damping but may reduce the life of the bearing. Heat generation in the rolling elements limits the speed therefore adequate lubrication must be supplied to all moving surfaces within the bearing. The effects of lubricants, especially oil, is very complex with few models providing a clear description of bearing failure. A high oil viscosity appears to increase the stiffness of the bearing but reduce damping [68]. Increases in rotational speed results in a general trend indicating a rise in stiffness, although the damping remains constant or decreases slightly. Increased oil flow for lubrication causes a corresponding increase of power losses but carries away more heat. The general recommendation is to provide enough oil to maintain an elasto-hydrodynamic film but no more. Heavy loads on the bearings may also increase heat generation and therefore limit the speed capability. If a belt drive is used, the belt tension may create heavy loads on the rear bearings and influence the spindle performance. Finally, the bearing torque or friction increases with the increased load due to greater deformation and resistance to the rolling elements, the bearing and retainer size, and increased drag at higher speeds.

The design considerations for a spindle-bearing system are numerous and characterized by general trends and empirical formulas used to describe poorly understood behavior. For these reasons the Spindle Design System presently concentrates on the preliminary design configuration of high-speed spindles. The interactive nature and explanation capabilities are used to promote an understanding of the complex nature of spindle and the many influential design parameters.

CHAPTER 5

MODULE DESCRIPTIONS AND DEVELOPMENT

The High-Speed Spindle Design System demonstrates a functional characteristic similar to the Design Process described as a Man-Machine System [33]. The spindle system illustrates the integrated definition of the product based on form and function. The design and analysis activities are responsible for the definition of the spindle form. Typically CAD/CAM approaches to design only address the product as a physical artifact designated by material, shape, tolerances, and assembly specifications, while essentially neglecting the product function. In the Knowledge-Based System described, the function of the spindle is determined through application descriptions of machining processes.

The Interactive Design System places the user at the center of the conceptual design process by aiding in the definition and representation of function requirements as well as by providing a means of design evaluation. Throughout the design process the user maintains the role of decision-maker while the system provides updated information. The system employs a design task model which performs and supports the tasks of the designer. The variational design system proposes

a spindle-bearing configuration based on selection rules, catalogue data, computational formulas, and past design data.

A hierarchical organization is utilized in order to fragment the process specification and preliminary design phase into a number of unique tasks of limited scope requiring only localized knowledge bases. Each group of tasks which perform a single independent function is combined to form a system module. A bottom-up integration of modules is used to develop the modules independently and maintain control flexibility. This modularity enables the user to access the system in a variety of fashions depending on the level of knowledge and design detail defined. Figure 5.1 illustrates a common calling sequence for an automated design process neglecting possible cycles. The calling sequence exhibits a clear representation of the seven stages of classical engineering design incorporated by the system methodology. The Roman numerals appearing to the right of the module names correspond to the following design stages described by Sandor [57]:

I Confrontation and Information Sources

II Formulation of Problem and Preparation of Information

III Selection of Design Concepts

IV Synthesis

V Analyzable Model

VI Experiment, Analysis, and Optimization

VII Presentation.

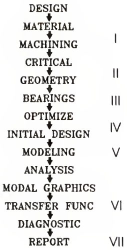


Figure 5.1 Modules

In order to efficiently code, debug, and maintain the Spindle program, a high level of cohesion exists while the level of coupling between the system modules is limited. A tool kit approach was employed in which a software library of generic program functions were evolved hierarchically to provide the more specialized functions of individual program modules. The system is written in ANSI standard FORTRAN, except for Assembly-based graphics, and consists of over 30 modules and subroutines.

5.1 Design Module

The Design module is the main calling and control module for the Spindle Design System. The module provides guidance for the initial installation process in which the system

hardware is defined and saved in the file INSTALL.DAT. The installation file contains three code values corresponding to variables PC, NSCREEN, and AR within the system program. During the initial program installation, PC is set either to 1, if the system is being run on a personal computer, or to 0, for all other hardware platforms. The platform type is determined so that the program may circumvent assembly-based graphics capabilities if a personal computer is not used. NSCREEN represents the graphics capability of the PC hardware where values of 6 and 16 correspond to CGA and EGA, respectively. Finally the screen aspect ratio is stored in AR. The aspect ratio refers to the ratio in which the screen displays vertical and horizontal images (AR = vert./horiz.). In order to determine the aspect ratio, a square is drawn on the screen. If the image appears square then the aspect ratio is 1. If the image appears to be a rectangle, the vertical and horizontal lengths are measured directly off the monitor by the user and the appropriate ratio may be calculated and entered. Typical aspect ratios are 2.25 for CGA and 1.25 for EGA graphics capabilities.

The Design module provides four primary modes of operation of the design program and allows the user to select either an English or Metric unit system. The menu format of the system interface is illustrated by the following examples. A computer-initiated dialogue is employed for novice users in the form of either menus or question-answer prompts.

CODE	OPTION	CODE	UNITS
0	System Information	0	English (lb,in)
1	Automated Design	1	Metric (N,mm)
2	Bearing Stiffness		
3	Model Design	Enter Units Code :	
4	Analysis		

Enter Option Code :

A brief description of the spindle design system and user instructions are provided by System Information. A fully interactive design session may be invoked through the choice of Automated Design. The user may perform a quick estimate of a bearing stiffness and determine the rolling element load distributions based on the geometry, preload, and applied loading of a specific bearing. If a conceptual design already exists, the user may describe the design interactively through Model Design or off-line for the purpose of modeling. Finally, a modal analysis of a specific spindle or design alternative may be initiated through the choice of Analysis. Each of these functions will be further explained within individual module descriptions.

The use of consistent units within the requested units system is aided by the appropriate units following all requests for input or displayed output values as shown below.

```
Enter Cutter Diameter (mm ) >
Spindle Power Required :  xxx.x  kW
```

The system can design both single purpose spindles such as those used in manufacturing transfer lines or multi-purpose spindles designed for a range of milling operations within machining centers. If the sum of the estimated usage entered by the designer of a multi-purpose spindle exceeds 100% the system will automatically normalize described operations at the user's request. In either instance, the design constraints and performance objectives of the spindle design are established from process information. In this way, the front end of the automated design process may be employed to more fully specify performance requirements for purchase specifications of a spindle or machine tool without continuing the conceptual design.

5.2 Material Module

The Material and Machining modules are used to confront the spindle design problem. Because the engineer usually lacks sufficient information to establish a complete problem statement including performance objectives and design constraints, the "real need" of the spindle is derived from the interactive description of application information and machining process parameters.

The Material module is used to determine the Unit Power and Cutting Stiffness properties of workpiece materials. If the user is sufficiently experienced or is using a material

not listed within the module, these values may be entered directly, otherwise they are determined interactively based on material hardness. The primary source of material property information for machining is the Machining Data Handbook, complied at the Machinability Data Center in Cincinnati, Ohio by MetCut Research Associates, Inc.

Due to the rapidly increasing number of tool/workpiece material combinations, provisions have been made to additionally implement rules concerning acceptable tool cutting edge materials. Subroutine Tool has been provided for such future development applications. The materials which are handled automatically by the Material module are listed below.

Code	Material
1	Aluminum Alloys
2	Cast Irons
3	Copper Alloys
4	High Temp Alloys (Ni, Co, Fe)
5	Steels (Plain C, alloys, tool)
6	Stainless Steel
7	Titanium

Enter Material Code :

When defining multi-purpose spindle applications the previously specified workpiece material may be accepted by default or an alternative material may be defined or selected by the user. Presently up to 10 different materials may be chosen, including those materials listed in the module as well

as the user defined materials. This limit may be expanded if needed but the program memory requirements increase accordingly.

5.3 Machining Module

The Machine module is used to define the milling operations required by the desired spindle and to determine cutting parameters necessary to calculate machine requirements. As Voelcker states [77,p.182], "The key to 'vertical' manufacturing-process automation seems to lie in finding effective computational models for processes (machining, . . .)." The critical limit of stability for regenerative chatter caused by the regeneration of waviness of the machined surface depends on the flexibility between the tool and workpiece. The Machine module prompts the user for the cutting process parameters including cutter diameter, number of teeth on the cutter, and the maximum spindle speed desired for each operation. In addition, chip load and axial depth of cut information is requested. The influence of the input parameters on the Cumulative Chip Width, B, and stability is described [62]. The Cumulative Chip Width may be expressed as

$$B = n\ b\ a/d$$

where

n = number of teeth on the cutter

b = axial depth of cut (chip width)

a/d = radial immersion (ratio of the radial width of cut over the cutter diameter).

The program assumes the worst case value of the radial immersion of a/d=1.0 corresponding to a slotting process. Although such an assumption may seem to be over constraining, the chance of entering improper information is eliminated. Geometric changes especially while cornering and pocketing often result in greater momentary radial immersions than specified by the user within manufacturing routines. A common example of this situation is illustrated by the four fluted endmill in Figure 5.2 where a specified 1/2 immersion cut actually requires full immersion of the cutter within the corner.

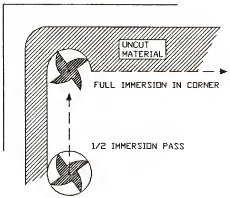


Figure 5.2 Toolpath in Corner

The required spindle stiffness is approximated by assuming a simplified single degree of freedom model of the milling process as shown in Figure 5.3. The value of the chip width (axial depth) may be used as a single process parameter by which the limit of stability for regenerative chatter in milling may be defined [69].

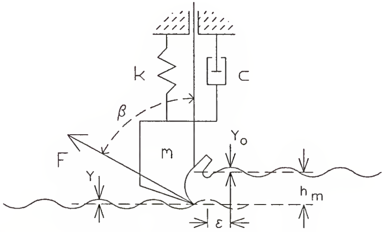


Figure 5.3 1 DOF Model of the Milling Process

The figure depicts the generation of the current surface designed by y, and the preceding pass defined by y_0,

$$y_0 = Y_0 \sin(\omega t)$$

$$y = Y \sin(\omega t + 2\pi N + \epsilon)$$

where N is the number of complete waves between subsequent passes and ϵ represents the phase difference between corresponding undulations such that $\epsilon/2\pi < 1$. Although a mean

chip thickness (radial depth), represented by h_m is defined by the feed per tooth or chip load, the actual chip thickness varies with resulting surfaces,

$$h = h_m + y_0 - y = h_m + h_v \ .$$

The actual chip thickness may be defined by the mean chip thickness and a variable chip thickness, h_v. In a similar way the average cutting force, F, may be defined by a mean force, F_m, and the variable force value F_v,

$$F = K_s \ b \ h = F_m + F_v \ .$$

The following assumptions are made during the modeling of the milling process:

- the vibratory system of the machine tool is linear
- the direction of F_v is constant
- F_v depends only on the vibration in the normal direction to the cut defined by y
- the value of F_v varies proportionally and instantaneously with h_v
- the frequency of the vibration and phase shift of undulations in subsequent, overlapping cuts are not influenced by the relationship of wavelength to the cut length (ie. the process is represented as a planing operation).

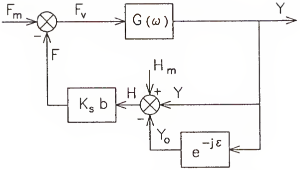


Figure 5.4 Model Block Diagram

Based on the assumptions, Figure 5.4 is a block diagram which represents the normal vibration, y, in regenerative cutting and implicitly includes the relationships described above. The oriented transfer function between the tool and the workpiece is represented by $G(\omega)$, where $Y=FG(\omega)$. At the limit of stability for self-excited, regenerative chatter, the magnitude of vibration of subsequent passes are equal, $|Y|=|Y_0|$, therefore the amplitude of vibration neither grows nor decays. The limited stability condition may be defined using the Nyquist Criterion, specifying that the open-loop transfer function of the system has a value of -1 as expressed by the following relationship,

$$K_s \; b \; G(\omega)(1-e^{-j\epsilon}) = -1 \quad .$$

The stability limit of the process may be represent by b_{lim},

$$b_{lim} = \frac{-\,1}{K_s \; G(1-e^{-j\epsilon})} \quad .$$

Because K_s and b are real, the limiting condition may be satisfied only if the expression $G(1-e^{-j\epsilon})$ is real. The vectors for subsequent undulations are shown in the complex plane by Figure 5.5. The phase difference is represented by the unit vector $e^{-j\epsilon}$, therefore the vector magnitudes must be equal. By inspection, the conditions of $|G|=|Ge^{-j\epsilon}|$, and $G(1-e^{-j\epsilon})$ are only maintained if $G(1-e^{-j\epsilon})=2Re(G)$.

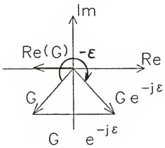


Figure 5.5 Complex Plane

The width of chip expressing the limit of stability is described by the equation,

$$b_{lim} = \frac{-1}{2 K_s Re[G]} .$$

If the phase shift is free to vary, a range of chatter limits signified by b_{lim} will occur at varying frequencies. The minimum value of the limit chip widths may be used to describe the absolute borderline for stability,

$$(b_{lim})_{min} = \frac{-1}{2\ K_s\ Re[G]_{min}}$$

where

K_s = cutting stiffness (unit power of workpiece matl.)

$Re[G]_{min}$ = minimum real part of the oriented transfer function between the tool and workpiece.

For instability to occur, the minimum part of the real transfer function must be negative as defined below,

$$Re[G]_{min} = \frac{-u}{4\ k\ \zeta\ (1+\zeta)}$$

where

k = the stiffness of the model in the normal direction

ζ = the damping ratio = c/c_{cr} = $c/(2\sqrt{km}\ )$

u = direction factor of average cutting force = $\cos\beta$

The minimum real value of the transfer function occurs at approximately $w_n(1+\zeta)$. If the workpiece is assumed to be rigid, then k corresponds to the dynamic stiffness measured at the cutting tool. Because the majority of system flexibilities influencing stability are frequently concentrated in the spindle, k may be conservatively used to set the minimum dynamic stiffness of the spindle. Assuming

a value for the directional factor and damping ratio, the spindle stiffness requirement may be calculated as

$$k = \frac{b\ K_s\ u}{2\zeta(1+\zeta)}\ .$$

Although the values may be changed within the module, a directional factor of u=0.5 is assumed. The damping ratio is set at ζ=0.03, (3%). The selection of the damping ratio at this time is subjective and may be updated based on a variety of information. In addition, consideration of process damping is not implemented in the module but may be added to a limited degree, or the user may at least be notified of the potential effect of additional damping due to the characteristics and geometry of a specific machining operation as described by Tlusty [69,70]. There are no provisions made in the module for taking advantage of favorable phasing of undulations in the machined surface or stability lobes.

Once the application descriptions and performance objectives are determined for each of the characteristic operations, the information is stored in the process data structure appropriately named DATA. The data structure location is defined by row corresponding to the application number, NPT, and the column where the information is stored. The following list describes the process data structure storage order and the variable meanings listed to the right of each entry.

DATA(NPT,1)=PERCENT	percentage of spindle usage
DATA(NPT,2)=MCODE	workpiece material code
DATA(NPT,3)=UP	unit power of workpiece material
DATA(NPT,4)=CS	cutting stiffness material
DATA(NPT,5)=IC	tool edge material index
DATA(NPT,6)=MILL	milling operation index
DATA(NPT,7)=CDIA	cutter diameter
DATA(NPT,8)=NTEETH	number of cutting edges on tool
DATA(NPT,9)=CUTLEN	cutter length
DATA(NPT,10)=CRHO	cutter material density
DATA(NPT,11)=EC	modulus of elasticity of cutter
DATA(NPT,12)=RPM	spindle speed in rev/min
DATA(NPT,13)=CLOAD	chip load
DATA(NPT,14)=DEPTH	depth of cut
DATA(NPT,15)=FEED	required feed rate
DATA(NPT,16)=MRR	material removal rate
DATA(NPT,17)=SPEED	surface machining speed
DATA(NPT,18)=POWER	delivered spindle power required
DATA(NPT,19)=STIFF	spindle stiffness required
DATA(NPT,20)=FREQHZ	tooth impact frequency in Hz

Based on the process application descriptions and the defined performance requirements, preliminary design requirements and alternatives may be evaluated.

5.4 Critical Module

The spindle design is formulated by the Critical and Geometry Modules which utilize the performance objectives and design requirements to explicitly define the problem. In addition, user-specified design constraints are established.

The machining operation which is critical to the design of the spindle is determined by subroutine Crit. The module is called only during the design of a multi-purpose spindle where up to 10 characteristic machining operations may be defined by the user. The Critical module calls the subroutine Ranker which sorts the indexes of the rows of the process data structure by values stored in a given column and creates a permutation vector corresponding to the ranked value indexes. Ranker also returns the minimum and maximum values of the sorted column. Through the ranking of the power, speed, and stiffness requirements of each machining operation the module assigns weighted values corresponding to machine utilization to each of the operations. The operation resulting with the greatest ranked value is used to define the critical requirements of the spindle design. The ranking procedure is used in order to restrict the program from over-designing the spindle based on a wide range of design parameters or the limited use of a highly demanding operation. In addition, the Crit module summarizes machining operation requirements and checks the parameter ranges to warn the designer of large variations which may indicate the need to distribute machining

operations among additional spindles or machining centers. The parameters by which the program continues the spindle design are then summarized for the user. If the designer wishes to override the module's assignment of the critical machining application, an alternative application number may be specified.

A brief example of three machining operations which were described by the user is outlined in Table 5.1. The Crit subroutine determined which of the operations was critically limiting to the spindle design and a section of the program output response is listed.

Table 5.1 Summary of Operation Requirements

Milling Operation	Workpiece Material	Power (kW)	Speed (RPM)	Stiffness (N/mm)	Utilization (%)
FACE	Cast Iron	11.4	1500	173,582	45
END	Aluminum	2.7	5300	79,699	45
END	Aluminum	15.4	7500	318,798	10

Program Output:

```
NUMBER OF OPERATIONS :  3
CRITICAL OPERATION # :  1

MIN SPINDLE SPEED    :  1500. RPM
MAX SPINDLE SPEED    :  7500. RPM

MIN POWER AT SPINDLE :  2.7   kW
MAX POWER AT SPINDLE :  15.4  kW

MIN SPINDLE STIFFNESS REQUIRED :  79699. N/mm
MAX SPINDLE STIFFNESS REQUIRED : 318798. N/mm
```

DESIGN PARAMETER SUMMARY 1

```
--------------------------------------
WORKPIECE MATERIAL   : Cast Iron
MACHINING OPERATION  : Face Milling
TOOL EDGE MATERIAL   : Coated Carbide
DIAMETER OF CUTTER   :     105.00 mm
NUMBER OF CUTTER TEETH :   8
MAX SPINDLE SPEED    :     1500. rpm
MAX CHIP LOAD        :      .150 mm/tooth
MAX DEPTH OF CUT     :     1.500 mm

AXIS FEED RATE REQ.  :     1.8 m/min
METAL REMOVAL RATE   :     284. cm^3/min
CUTTING SPEED        :     495. m/min
SPINDLE POWER REQ.   :     11.4 kW
SPINDLE STIFFNESS REQ. :   173582 N/mm
TOOTH IMPACT FREQ.   :     200. Hz
```

5.5 Geometry Module

Subroutine Geo is used to specify the form of the design by determining geometric constraints and requirements of the spindle. The user must specify a standard spindle nose size (30,40,45,50) in order to define the gage diameter of the spindle taper. From this information, dimensional ranges of the inner and outer spindle diameters, bearing span, and overhang distances are defined. The user may constrain the minimum and maximum overall spindle length or the overhang of the spindle nose face from the front bearing. The spindle overhang is defined by the distance from the front bearing to the spindle nose face. Utilizing geometry rules, the program checks the validity of user constraints and imposes them on

the spindle design. The designer may also define the geometry of a cylindrical tool extension to be used in conjunction with the spindle and tool. The extension will be included in the modal analysis of the spindle configuration.

5.6 Bearings Module

Through the use of the Bearings module an appropriate spindle-bearing configuration is determined based on production rules. The module seeks to encode the experience of skillful designers in a manner useful for transferring such knowledge to novice designers. Figure 5.6 illustrates the basic spindle configurations from which the program selects an initial configuration. Each of the configurations is based on the use of rolling bearings [78]. Several advantages and reasons for utilizing rolling bearings are listed below:

- high level of standardization and interchangeability
- high load capacity even without rotation
- allows high fluctuations of load and rotational speeds
- low power loss and minimal friction
- small width space requirements.

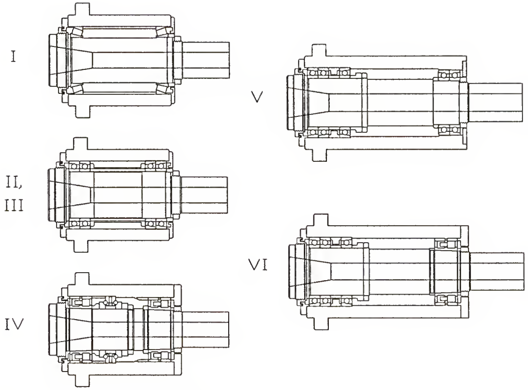


Figure 5.6 Spindle Configurations

The initial spindle configuration is chosen based on geometric requirements and constraints, the loading of the spindle, operating speed ranges and stiffness requirements. An example of a production rule which illustrates a Forward Chaining reasoning mechanism is shown below. Forward Chaining starts with an initial state and then finds a fact whose premise is verified by the state. The process is repeated until one top-level goal conclusion is reached, [80]. In this example the state defining the spindle characteristics results in the conclusion of a permissible spindle configuration.

IF the spindle is SHORT

AND the loading is predominantly RADIAL

AND the speed range is LOW

THEN a TAPER ROLLER BEARING configuration is appropriate.

After the initial spindle configuration is determined, the Bearing module estimates the stiffnesses of the front and rear bearing groups. Because the stiffness of a rolling bearing mounted without clearance is a nonlinear function of the bearing geometry and the external force applied,

$$F = K \delta^q$$

where

F = normal force between rolling element and raceway

K = load deflection coefficient for ball or roller contact

δ = deformation

q = 3/2 for point contact, 10/9 for line contact,

the calculation of the bearing stiffnesses may be greatly simplified by the use of equations founded on empirical data. If the user desires a more complete description of the load-deflection characteristics of a bearing, including the calculation of the statically distributed (steady state) load on each of the roller elements, the user may interactively

analyze a bearing. In subroutines Radial and Combined the classical Hertzian solution [26] is utilized although the calculation of the ellipticity parameter k_e, and the complete elliptic integrals of the first (ϵ_1) and second (ϵ_2) kinds are simplified by the least squares expressions developed by Brewe and Hamrock [25]. The curve fitting equations which follow introduce less than 2% error while significantly reducing the computational requirements,

$$k_e = 1.0339\ (R_y/R_x)^{0.6360}$$

$$\epsilon_1 = 1.5277 + 0.6023\ \ln(R_y/R_x)$$

$$\epsilon_2 = 1.0003 + 0.5968/(R_y/R_x)$$

where

R_x = effective radius in the X direction

R_y = effective radius in the Y direction

and R_y and R_x are defined by the curvature sums,

$$(1/R_x) = (1/r_{ax}) + (1/r_{bx})$$

$$(1/R_y) = (1/r_{ay}) + (1/r_{by}).$$

The load deflection constant for the contact may then be calculated by,

$$K = \pi\ k_e\ E' \left[\frac{R\ \epsilon_2}{4.5\ \epsilon_1^{\,3}}\right]^{1/2}$$

where R is the curvature sum defined by R_y and R_x and E' is the effective elastic modulus of the contacting bodies.

For cylindrical rolling elements in which a line contact is formed and the resulting contact area is a rectangle instead of an ellipse, empirical data is used to predict the force-deflection relationship. The following equation has been experimentally verified and indicates that the load deflection coefficient is solely dependent on the effective contact length of the roller, [18]. This relationship is used within the bearing analysis subroutines for line contact,

$$F = 5.66\text{x}10^4 \; L_e^{0.9} \; \delta^{10/9} \;,$$

where

F = normal force between roller and raceway (N)

L_e = effective contact length of the roller (mm)

δ = deformation (mm).

Founded on fundamental assumptions and the application of classical theory of elasticity, the ball-outer-race contact illustrated in Figure 5.7 may be defined. The following assumptions are made,

- the materials are homogeneous
- the yield stress of the materials is not exceeded
- the solids are at rest and in equilibrium (steady state)
- no tangential forces are induced between the solids
- the contact is limited to a small portion of the surface.

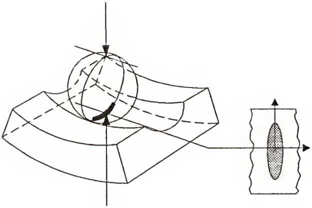


Figure 5.7 Ball/Outer-Race Contact

Based on the degree of geometric conformity, the contact parameters expressing the ellipticity, the applied normal force, Poisson's ratio, and the effective modulus of elasticity of the contacting solids, the semimajor and semiminor axes of the contact ellipse, a and b respectively, and maximum deformation at the center of the contact (δ_{max}) can be calculated by the following equations [25],

$$a = \left[\frac{6\ k_e^2\ \epsilon_2\ F\ R}{\pi\ E'}\right]^{1/3}$$

$$b = \left[\frac{6\ \epsilon_2\ F\ R}{\pi\ k_e\ E'}\right]^{1/3}$$

$$\delta_{max} = \epsilon_1 \left\{ \frac{9}{2\ \epsilon_2\ R} \left[\frac{F}{\pi\ k_e\ E'}\right]^2 \right\}^{1/3} .$$

The loads exerted on a bearing are transmitted through the contacts between the rolling elements and the raceways. The geometry of the rolling elements and their material properties determines how well the bearing can transmit the loads during operation. Heat generation and power losses are also a function of the contact geometry and are primarily dependent of the ball and race groove conformity.

The total approach of the two raceways for any rolling element under load is the sum of the approaches at each element/raceway contact,

$$\delta_t = \delta_i + \delta_o$$

where

δ_t = total approach for any element

δ_i = approach at the inner race contact

δ_o = approach at the outer race contact.

By including the summed deformations calculated at both the inner-race-ball contacts and the ball-outer-race contacts, the load distribution on the balls or rollers may be expressed and the bearing stiffness may be approximated. The outer race is assumed to be fixed relative to the spindle housing. The program assumes both the races and the rolling elements are steel although fully-commented equations are provided within the code to the modification of the materials and their properties if required.

5.6.1 Radial

Subroutine Radial is used to estimate the stiffness of radial ball bearings and cylindrical roller bearings with diametral interference subjected to an externally applied radial load. During the design process either a quick estimate of the radial stiffness may be made based on heuristics or a more complete analysis of the bearing may be selected.

The complete analysis includes the calculation of the element load distribution as well as the radial deflection and stiffness of deep groove radial ball bearings and cylindrical roller bearings. The subroutine assumes a constant element contact angle of zero, such that β=0.

The deflection of the θ_ith angular position is given the following expression,

$$\delta\theta_i = \delta_r \cos\theta_i + \tfrac{1}{2}P_d$$

where

$i = 1,2,\ldots,NE$

NE = the number of rolling elements in a single bearing row

δ_r = radial deflection of inner bearing race

P_d = diametral interference providing preload.

The normal force on the element at the angular position corresponding to θ_i is determined by the relation,

$$F\theta_i = K\ \delta\theta_i^{\,q}.$$

In order for force equilibrium to be satisfied, the magnitude of the sum of the internal radial force components must equal the magnitude of the externally applied radial load (F_r),

$$F_r = \sum_{i=1}^{NE} F\theta_i \cos\theta_i = \sum_{i=1}^{NE} K\ (\delta_r \cos\theta_i + \tfrac{1}{2}P_d)^q \cos\theta_i$$

subject to the constraint and nonlinearity that if $\delta\theta_i<0$ then $F\theta_i=0$. The total radial deflection (δ_r) must be solved in order to balance any given external radial load. Assuming an initial δ_r, the deflection and load for each rolling element can be computed and the error associated with the force balance represented by R,

$$R = F_r - \sum_{i=1}^{NE} K\ (\delta_r \cos\theta_i + \tfrac{1}{2}P_d)^q \cos\theta_i\ .$$

To solve for δ_r, the Newton-Raphson Method is employed for determining the roots of the equation R=0. The method is based on the iterative procedure,

$$\delta_r^{j+1} = \delta_r^{j} - \frac{R(\delta_r^{j})}{R'(\delta_r^{j})}$$

where

j = the iteration number for δ_r

$R(\delta_r^{j})$ = the error function based on the j^{th} iteration of δ_r

$R'(\delta_r^{j})$ = the derivative of $R(\delta_r^{j})$ with respect to δ_r.

The derivative of R with respect to δ_r is given by the following equation,

$$R' = \frac{d\,R}{d\,\delta_r} = -\sum_{i=1}^{NE} q\,K\,(\delta_r \cos\theta_i + \tfrac{1}{2}P_d)^{q-1} \cos^2\theta_i$$

The iteration process continues until the correction factor, R/R', is less that 0.1% of the calculated radial deflection. The tolerance limit allows accurate calculation of the element load distributions and bearing radial deflection typically in less than five iterations.

The following is an example of an interactive session within Radial. The user responses are the input values to the right of the prompts (:). Note that although a numerical value for the applied load is requested, the user may enter a ? to ask the system why the data is required. This feature is particularly helpful for novice users. In addition, typical values or rules are offered with an explanation

especially when the information requested that is not commonly known to an inexperienced designer. The user may invoke such help facilities by replying to the system prompt with H.

Program Output:

Code	Rolling Element Type
0	Ball (point contact)
1	Roller (line contact)

Enter Rolling Element Type Code : 1

Enter the effective length of a roller (mm) : 12.6

Enter the number of bearing rows : 2

Enter the number of rolling elements per row : 28

Enter the diametral preload (mm) : H

Although diametral clearance (preload≤0) is used with single row radial bearings, for use in spindles diametral interference is used to preload the bearings.

A typical preload value would be 0.012 mm.

Enter the diametral preload (mm) : 0.012

Enter the external radial force (N) : ?

The load deflection relationship of a bearing is non-linear as shown below,

$$F = K\,\delta^{q}$$

where

F = externally applied radial force (N)
K = calculated load deflection coefficient
q = 3/2 for point and 10/9 for line contact

An external force > 1 (N) must be entered in order to calculate the load distribution and bearing stiffness. A radial force value typically exerted on the bearing during spindle use will yield the most accurate results.

```
Enter the external radial force (N ) : 10000

*********************  Be prepared to stop the results
* Analysis complete *  from scrolling past the bottom
*********************  of the screen.


RESULTS:          ITERATION =          3

    J      ANGLE     DEFLECTION      FORCE
 ---------------------------------------
    1       13.      .8319E-02       2336.
    2       26.      .8143E-02       2282.
    3       39.      .7859E-02       2194.
    4       51.      .7483E-02       2077.
    5       64.      .7032E-02       1938.
    6       77.      .6529E-02       1785.
    7       90.      .6000E-02       1625.
    8      103.      .5471E-02       1467.
    9      116.      .4968E-02       1318.
   10      129.      .4517E-02       1185.
   11      141.      .4141E-02       1076.
   12      154.      .3857E-02        995.
   13      167.      .3681E-02        944.
   14      180.      .3622E-02        927.
   15      193.      .3681E-02        944.
   16      206.      .3857E-02        995.
   17      219.      .4141E-02       1076.
   18      231.      .4517E-02       1185.
   19      244.      .4968E-02       1318.
   20      257.      .5471E-02       1467.
   21      270.      .6000E-02       1625.
   22      283.      .6529E-02       1785.
   23      296.      .7032E-02       1938.
   24      309.      .7483E-02       2077.
   25      321.      .7859E-02       2194.
   26      334.      .8143E-02       2282.
   27      347.      .8319E-02       2336.
   28      360.      .8378E-02       2355.

RADIAL DEFLECTION =    .2378E-02 mm
RADIAL STIFFNESS  =    .4187E+07 N/mm
```

The data input in the previous example session corresponds to an SKF NN3024 double row cylindrical roller bearing with a bore diameter of 120 mm. The bearing behavior may be further analyzed with successive runs of Radial. Figure 5.8

illustrated the load distribution on the rollers with the extreme values indicated in Newtons on the diagram. The three cases depicted on the right of the figure represent the magnitude of the forces distributed among the rolling elements and should not be confused with the position of either the inner or outer race. The load conditions correspond to no external load (the indicated load distribution is solely due to the preload), an applied load of 10,000 N, and a load of 25,000 N which fully relieves the preload resulting in a decreased radial stiffness.

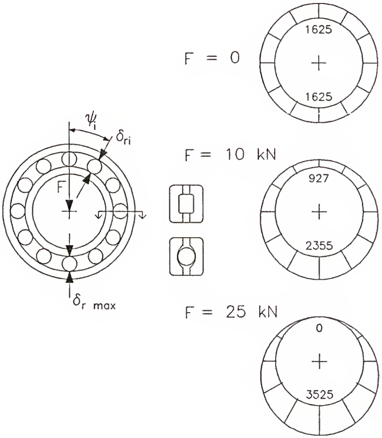


Figure 5.8 Analysis of an SKF NN3024 Bearing

For the 10 kN loading case, the relative accuracy of the simplified bearing analysis may be compared to the values calculated by the complete bearing analysis programs of SKF examples of which are shown in SKF [61]. If a quick bearing stiffness is requested the following rule would be used to estimate the radial stiffness [22],

$$K_r = 38500\ |\delta_{rp}|^{1/9}\ D_b^{10/9}$$

where

K_r = radial stiffness (N/mm)

δ_{rp} = radial preload (mm)

D_b = bore diameter of bearing (mm),

and the estimated stiffness would be K_r = 4.45E6 N/mm. Based on the calculations illustrated by Radial, the radial deflection of the bearing is 0.00238 mm and K_r = 4.187E6 N/mm. The results of SKF's bearing analysis programs yield a resulting deflection of 0.00244 mm and a radial stiffness of K_r = 4.098E6 N/mm. Assuming the results of SKF accurately reflect the bearing behavior the stiffness estimated by the quick method is 8.7% high. The more complete analysis of RADIAL which assumes no bearing misalignment and is based on the curve-fit expressions of the elliptic integrals is 2.1% high. For the purposes of comparison and evaluation of preliminary designs the accuracy of these results is quite reasonable.

5.6.2 Combined

For both a radial and axial loading of an angular contact bearing, subroutine Combined is used. In Combined the races are constrained to relative motion in parallel planes by the assumption that negligible misalignment of the bearing occurs. The calculated displacements are limited to a radial displacement δ_r and an axial displacement δ_a from which the bearing stiffnesses are resolved. Because both constant force and constant distance methods are used to preload bearings, subroutine Combined allows the user to specify either type of preload information. In the analysis of angular contact ball bearings (ACBB), the change in the bearing free contact angle (β_f), based on the applied preload is determined by the program. In addition, the final contact angle resulting from the application of radial and axial external loads is reported in the bearing analysis summary. Figure 5.9 illustrates the geometry of a preloaded ball bearing [25]. If the preload is specified by a fixed distance δ_{ap}, the resulting contact angle may be determined directly, otherwise the preload force is used to iteratively determine the new contact angle (β).

If the preload of the ACBB is specified implicitly by the axial preload displacement (δ_{ap}), the contact angle and the resulting axial force due to the preload (F_{ap}), may be directly resolved. By inspection of the enlarged geometry

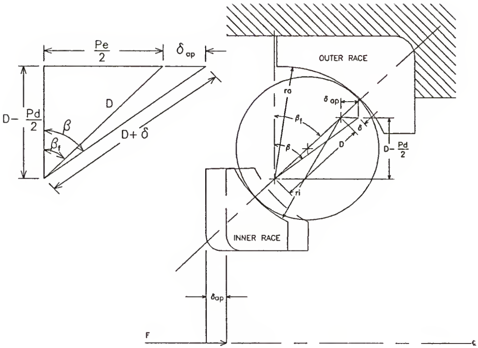


Figure 5.9 Angular Contact Ball Bearing Under Preload

shown to the left of the ball bearing in Figure 5.9, the overall lengthdefined by the endplay ($\frac{1}{2}P_e$) and the axial preload may be equated by the following

$$D \sin\beta_f + \delta_{ap} = (D + \delta) \sin\beta$$

where the distance D, is defined by the conformity ratios of the inner and outer races, f_i and f_o respectively, and the bearing ball diameter (d),

$$D = (f_i + f_c - 1) d .$$

The free contact angle (β_f), may be related to the contact angle β, resulting from the preload displacement, $\delta=P_d$, by the equation,

$$D \cos\beta_f = (D + \delta) \cos\beta.$$

The contact angle initially resulting from the application of the preload may be calculated from the geometric relations discussed by,

$$\beta = \mathrm{Tan}^{-1} \left[\frac{\delta_{ap}/D + \sin\beta_f}{\cos\beta} \right] .$$

The corresponding axial preload force may be determined from the element loads which are equally distributed due to the uniform axial displacement. The axial force required to apply the specified δ_{ap} is determined by,

$$F_{ap} = NE\ (K\delta^q)\ \sin\beta\ .$$

If the preload of ACBB is specified by the axially applied preload force, the balance of the normal loads on the balls is calculated by the Newton-Raphson Method summarized by the following. The initial solution for β is estimated by $\beta=1.1\beta_f$, and the normal preload deflection is determined by the following relationship,

$$\delta = D\ (\cos\beta_f/\cos\beta - 1).$$

The error function for the normal deflection of the contact angle is expressed by the function,

$$F(\beta) = F_{\beta} = \frac{F_{ap}}{(NE \cdot K)} - \sin\beta \; D^{q} \; (\cos\beta_{f}/\cos\beta - 1) \; .$$

The derivative of the error function with respect to β is,

$$F_{\beta}' = -\cos\beta \; d^{q}(\cos\beta_{f}/\cos\beta - 1)^{q} - q\cos\beta_{f} \; \tan^{2}\beta \; D^{q}(\cos\beta_{f}/\cos - 1)^{q-1} .$$

The iterative process by which the contact angle is determine is summarized by

$$\beta^{j+1} = \beta^{j} - F_{\beta}/F_{\beta}' \; .$$

The iteration proceeds until the tolerance condition is reached such that

$$\frac{F_{\beta}/F_{\beta}'}{\beta} \leq 0.001 \quad .$$

The calculations are greatly simplified if roller bearings are employed because the contact angle remains constant (ie. $\beta_{f}=\beta$). Figure 5.10 illustrates a tapered roller bearing subject to an axial preload force. Both the normal force and the deflection may be directly calculated by the equations below,

$$\delta = \delta_{ap} \sin\beta \qquad \text{and} \qquad F = (F_{ap}/NE)\sin\beta \; .$$

With simple manipulation either δ_{ap} of F_{ap} may be determined from the user specified preload information.

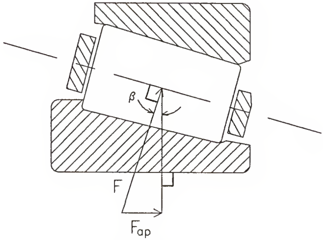


Figure 5.10 Taper Roller Bearing

In order to compute the distributed normal load and deflections of the rolling elements, the overall axial and radial deflections of the bearing, δ_a and δ_r respectively, must be solved simultaneously. Again, the Newton-Raphson Method is used to find the roots of the error equations,

$$A = F_a - \sum_{i=1}^{NE} F\theta_i \sin\beta$$

$$R = F_r - \sum_{i=1}^{NE} F\theta_i \cos\theta_i \cos\beta$$

where

A = the axial force balance error

R = the radial force balance error

β = the loaded contact angle .

The complete derivative of the equations for the loaded contact angle as well as the error derivatives are shown in Appendix A. The derivatives for the axial error (A'), and the radial error (R'), are summarized by the following,

$$A' = \sum_{i=1}^{NE} q\ F\theta_i/\delta\theta_i\ \sin^2\beta + F\theta_i\ \cos^2\beta/(D\cdot DEN)$$

$$R' = \sum_{i=1}^{NE} [q\ F\theta_i/\delta\theta_i\ \cos^2\beta\ \cos^2\theta_i + F\theta\ \cos^2\theta\ \sin^2\beta/(D\cdot DEN)]$$

where

$$DEN = \left[(\cos\beta_f + \delta_r/D\ \cos\theta_i)^2 + (\sin\beta_f + \delta_a/D)^2\right]^{1/2} = D + \delta\theta_i.$$

The Newton-Raphson Method results in rapid convergence (ie. usually less than 5 iterations), despite the fact that the cross correction terms are neglected.

The results of an analysis of a Fafnir 2MM 9120 angular contact ball bearing preloaded by a specified axial displacement follow. The results compare very closely with an analysis of the corresponding front bearing of the Sundstrand Series 20 Omnimil [29].

The geometric specifications of the bearing is followed by the calculated load-deflection constant (K), and the computed preload information. In this example, a fixed axial displacement was specified and the corresponding axial preload force was determined by the program. The loaded contact angle increased by nearly 25%, from 15° to 18.7° as shown in the

output. Essentially no external radial and axial forces are applied in this example therefore the analysis corresponds to the initially preloaded bearing and represents the greatest axial and radial stiffness that can be expected from the bearing. The 1 N radial force is specified only to assure a rapid balance of forces by the iterative procedure. The following is the bearing analysis written to an output file named by the user.

Program Output:

```
BEARING ANALYSIS SUMMARY         FILE : BEARING.OUT

BEARING : FAFNIR 2MM 9120

INPUT DATA   (ALL DIMENSIONS IN mm  )
************************************************

 ELEMENT     # OF     # ROLLING     CONTACT
  TYPE       ROWS     ELEMENTS      ANGLE(°)
----------------------------------------
 BALL         1         22            15.0


    BALL     RACE (GROOVE)    INNER      OUTER
    DIA.        RADIUS      RACE DIA   RACE DIA
-----------------------------------------------
 15.5000       8.0600       109.5000   140.5000


 LOAD-DEFLECTION CONSTANT (N/mm ^1.50)
-------------------------------------
             .3982E+06
```

AXIAL PRELOAD FORCE (N)	AXIAL PRELOAD DISP	LOADED CONTACT ANGLE (°)
1297.4	.02300	18.7

EXTERNAL RADIAL FORCE (N)	EXTERNAL AXIAL FORCE (N)
1.0	.0

ANALYSIS RESULTS

****

RESULTS: ITERATION = 4

J	ANGLE	DEFLECTION	FORCE
1	16.	.5971E-02	184.
2	33.	.5971E-02	184.
3	49.	.5970E-02	184.
4	65.	.5970E-02	184.
5	82.	.5969E-02	184.
6	98.	.5969E-02	184.
7	115.	.5968E-02	184.
8	131.	.5968E-02	184.
9	147.	.5967E-02	184.
10	164.	.5967E-02	184.
11	180.	.5967E-02	184.
12	196.	.5967E-02	184.
13	213.	.5967E-02	184.
14	229.	.5968E-02	184.
15	245.	.5968E-02	184.
16	262.	.5969E-02	184.
17	278.	.5969E-02	184.
18	295.	.5970E-02	184.
19	311.	.5970E-02	184.
20	327.	.5971E-02	184.
21	344.	.5971E-02	184.
22	360.	.5971E-02	184.

AXIAL DEFLECTION = .1944E-01 mm
AXIAL STIFFNESS = .1105E+06 N/mm

RADIAL DEFLECTION = .2201E-05 mm
RADIAL STIFFNESS = .4555E+06 N/mm

5.7 Optimize Module

The Optimization module employs the subroutine SOP (Simplex OPtimization), to determine the optimum spindle overhang and bearing span. The Simplex Method is based on a geometric figure consisting of N+1 points or vertices interconnected by line segments or edges in N dimensions [49]. For the optimization of 2 parameters (N=2) the simplex is a triangle. The geometric algorithm locates optimum combinations of bearing spacing values through an automated iterative search. The subroutine minimizes the continuous system function representing the spindle flexibility. Several advantages of the Simplex Method are listed:

- the algorithm requires only function evaluations;
- the function being evaluated need not be differentiable;
- the program is faster than brute force techniques, and the investigation of several variables simultaneously does not cause significant slowdowns;
- the program requires little memory (an order of N^2).

The system equations are set up to perform as a function with one unambiguous value resulting from a the set of input variables. The value of any variable is steered away from an unusable range defined by inequality constraints with an internal penalty function. The penalty function artificially

inflates the resultant therefore returning the vertex within the boundary condition. The difference between the best and worst point resultants establishes the termination criteria.

Figure 5.11 illustrates the general procedure by which the program plots values in N-dimensional space with each variable and a resultant dimension. One vertex of the polygon is discarded and replaced by a better one with each iterative cycle. The polygon changes size and shape according to the terrain, moving fastest in an area of uniform slope. As an optimum is approached, the resultant function evaluations described by the vertices become nearly equal.

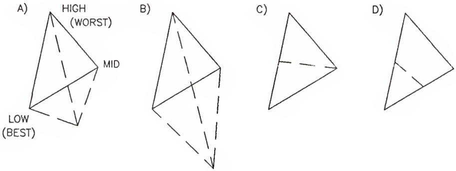


Figure 5.11 Geometric Simplex

The program identifies the best and worst value points, discards the worst, and calculates the midpoints of the remaining edges. Extrapolating from the worst point through the midpoint of the facing edge by a distance determined by the segment length, a reflected point is generated as in case A). An extended point is established with the addition of the

projected distance, B). If one of the two reflections represents an improvement, then the better one is selected as the new vertex. If neither reflection provides a better alternative, a contracted vertex is explored. The contracted vertex is placed midway between the high point and the low point, C). If still no improvement is reached, the worst and mid points are moved half their distance toward the best point as shown in case D). An appropriate sequence of such steps is used to converge to a minimum of the flexibility function. An alternative weighting method may be utilized in order to accelerate convergence if required by specifying the reflection point by the average of the polygon vertices as illustrated below in Figure 5.12.

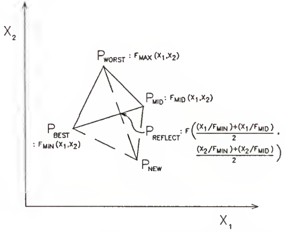
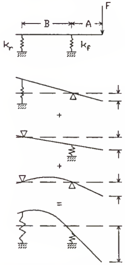


Figure 5.12 Weighting Method Accelerator

The module determines the static optimum of the bearing centers in a spindle configuration in which the front and rear bearing groups are represented by a two radial stiffnesses k_f and k_r, respectively. Constant cross-sectional properties

over the spindle length are applied. The total static deflection is comprised of a number of contributory elements. Bearing flexure, spindle flexure, and housing flexure are often significant [79]. For the Simplex Optimization the housing is assumed rigid and the deflection of the spindle is represented in Figure 5.13. The small triangles in the figure symbolize self-aligning bearings which are held fixed radially. The total deflection of a spindle subjected to an external force F applied at the spindle nose is determined by the deflections due to the bearings δ_B, and the spindle flexure δ_S. The module minimizes the spindle flexibility calculated at the spindle nose (δ_T/F) and returns optimum overhang A, and bearing span B, distances. A variety of nomograms, simplified equations, and design charts are published but the results may vary greatly and few accurately yield a static optimum.



$$\delta_B = \delta_1 + \delta_2 = \frac{F}{B^2}\left[\frac{(A+B)^2}{k_f} + \frac{A^2}{k_r}\right]$$

$$\delta_1 = \frac{F\,A^2}{k_r B^2}$$

$$\delta_2 = \frac{F\,A^2}{k_f B^2} + \frac{2\,A\,F}{B\,k_f} + \frac{F}{k_f}$$

$$\delta_3 = \frac{F\,A^2}{3\,E\,I}(A + B) = \delta_S$$

$$\delta_T = \delta_1 + \delta_2 + \delta_3 = \delta_B + \delta_S$$

Figure 5.13 Static Deflection of Spindle-Bearing System

An example case taken from Weck [79], in which a nomogram does accurately determine an optimum bearing spacing is outlined by the following example. The result from a simplified equation derived by SKF [61], is easily calculated although not as accurate,

$$B_{opt} = [6EI(1/k_f + 1/k_r)]^{1/3} .$$

The example parameters are given:

A = 350 mm (spindle overhang)

D = 100 mm (spindle diameter)

E = 21E4 N/mm^2 (modulus of elasticity for steel)

k_f = 10E5 N/mm (front spring rate)

k_r = 6E5 N/mm (rear spring rate)

The optimum bearing spacing B_{opt}, calculated by both methods are shown in Table 5.2 and compared to the "true optimum" determined within the Optimize module by the Simplex Optimization method. Although the optimum bearing spacing results vary significantly, as exhibited by the design curves in [60], the spindle stiffness may change nominally with a change in the bearing spacing.

Table 5.2, Optimum Bearing Spacing

Method	B_{opt}	Overall spindle stiffness
Nomogram [64]	290 mm	3.062 E4 N/mm
Equation [55]	254 mm	3.052 E4 N/mm
SOP	278 mm	3.065 E4 N/mm

5.8 Initial Design Module

The initial spindle configuration is graphically represented by the Gspindle module. The type of graphics monitor, aspect ratio, and resolution are known based on the initial program installation. The conceptual design is dynamically scaled for graphics purposes and parametrically assembled on the screen. A series of graphics subroutines which draw bearings, cylinders, and spacers support the Gspindle module. In return, the graphics subroutines call primitive subroutines written in Assembler language which are supplied by GRAPHICS.LIB. Additional development of the Spindle Design System may include standard graphics libraries such as PHIGS, XWINDOWS, or GKS.

5.9 Preliminary Spindle Modeling Module

The Preliminary Spindle Modeling module consists of two alternative subroutines SDAT and BDAT. SDAT is invoked during an automated spindle design and is transparent to the designer. The routine prepares a formatted spindle description which is stores in a disk file named BEAM.DAT. If the designer has already developed a conceptual design, BDAT may be directly called by the Design module in response to the user's choice of Option Code 3, Model Design.

BDAT interactively determines the unit system desired, tool and spindle materials, and the spindle geometry. The program will allow the addition of concentrated springs and masses to the spindle design. If the radial spring rate of a bearing is not known, the user may re-enter the design program and request Option Code 2, Bearing Stiffness. If angular contact ball bearings are used, the user may enter the coordinate location of each bearing and enter 0 for the radial stiffness. In response to the 0 stiffness input, the subroutine will estimate the spring rate based solely on the bearing bore diameter and inform the designer the assumed bearing stiffness. Finally, BDAT prepares the spindle description file just as SDAT. A more experienced user may also prepare a spindle description file outside of the design program and enter directly into the Analysis option. In addition, the spindle housing may be included in a simplified form in the description file.

Both SDAT and BDAT results in the generation of the file BEAM.DAT. The data file includes a formatted description of the spindle-bearing system. The format of the file is summarized below,

```
NE,NDOF,NCM,MD,NCK,MUNIT
I,MCODE,XTYPE,SL,D1,D2,N1,N2,N3,N4
USMRHO,USME
JC,CM
JC,CK,JC2
```

where the variables are defined by the following:

NE	=	number of beam elements in model
NDOF	=	total number of degrees of freedom
NCM	=	number of concentrated masses
MD	=	mass distribution (0=lumped, 1=continuous)
NCK	=	number of concentrated springs (bearings)
MUNIT	=	unit code (0=English, 1=Metric)
I	=	segment numbers
MCODE	=	material code identifier (0-3)
XTYPE	=	cross section type (1=cir., 2=rectangular)
SL	=	segment length
D1,D2	=	characteristic dimensions (OD,ID or W,H)
N1-N4	=	coordinate numbers (N=0 implies fixed) N1,N3 transnational, N2,N4 rotational
USMRHO	=	user defined material density (if MUNIT=3)
USME	=	user defined modulus of elasticity
JC,CM	=	coordinate location and concentrated mass
JC,CK,JC2	=	end location, concentrated spring rate, other end coordinate location

The total number of lines of formatted input in the description file depends on the number of segments and the number of user-defined materials. The USMRHO, USME line is only inserted directly after segment information where the material code is 3. The number of concentrated mass and spring data must correspond to the NCM and NCK, respectively.

5.10 Spindle Modeling Module

The Spindle Modeling module consists of the subroutine Beammod which reads the file BEAM.DAT. Based on the spindle description a discretized representation of the continuous spindle system is represented by Finite Element Modeling, (FEM). The principle idea behind the Finite Element Method is disarmingly straight forward. Instead of defining the admissible governing functions over the entire global domain, the functions are defined only over the relatively small subdomains called "finite elements." The method was first suggested by R. Courant in what has come to be regarded as a classic paper [8], although the term "finite element" was coined many years later in a paper by R.W. Clough in 1960 [7]. Today the concept of Finite Element Modeling, is a broad one. A widely used formulation for problems in the analysis of structural and solid mechanics is the displacement-based finite element method.

The discretization of the structure or mechanical system involves determining the number, type, arrangement, and distribution of finite elements in the problem domain. Proper specification of the node locations and element shapes is essential in order that the discrete spindle model represents the actual continuous system as close as practically possible.

As discussed in Optiz and Noppen [45], the stiffness and mass properties of the components are of primary interest for

the design of machine tool structures. For the purpose of calculation, the spindle is subdivided into a number of structural elements. The elements are connected to each other at nodal points each of which has associated degrees of freedom. By superposing the load-deflection relationships for all of the elements of the model, the nodal forces and resulting displacement may be related by a global stiffness matrix. The superposition of the element stiffness matrices is carried out in such a way that the internal and external loads at every node are in a state of equilibrium. A set of simultaneous linear equations may be generated assuming a static, linear analysis and accounting for the boundary conditions and external loads acting on the structure.

The proper idealization of the spindle is the most important step in FEM and refers to selecting the types of finite elements. The element's type is characterized by the properties (truss, beam, membrane, plate), the order of representation (linear, quadratic), and the shapes (triangular, quadrilateral, brick). Guidelines for the establishment of a FEM are reviewed in [65]. The optimum element choice is dependent on geometry, capabilities, and limitations of the element as well its ability to yield accurate results with minimal computational effort. Finally the set of dynamic degrees of freedom (DOF), must be specified. The following general rules of thumb are described in [63] and were used to describe the behavior of the spindle within the automated design process:

- the number of dynamic DOF should be at least twice the highest mode of interest;
- for modal analysis, the number of reduced modes equals the number of dynamic DOF, but only the "bottom half" of the calculated modes should be considered accurate;
- dynamic DOF should be placed in areas of large mass and low rigidity for these areas tend to drive the mode;
- distribution of the DOF should effectively describe the anticipated mode shapes;
- DOF must be selected at each point of force application;
- DOF do not have to be selected at permanently constrained points.

A proper finite element solution should converge to the analytical, exact solution of the differential equations that govern the response of the spindle idealization. To assure monotonic convergence the elements must be complete and compatible. If these conditions are fulfilled, the accuracy of the solution results increases continuously as the number of elements grows. The requirement of completeness of an element implies that the displacement functions of the element are able to represent the rigid-body displacements and the constant strain states. The finite element model and analysis may result in errors due to the approximation by discretization, solution of the dynamic equilibrium state, solution of the finite element equations by iteration, and round-off.

5.10.1 Beam Elements

The objective of the spindle model is to predict the natural frequencies and mode shapes of the preliminary spindle design. The global model of the spindle-bearing system must contain sufficient detail to characterize the stiffness and mass of the structure. Beam elements work well in applications where only dynamic displacement responses are required and good results are obtained from a relatively course model with few elements. The requirement of compatibility, which assures that displacement within elements and across the element boundaries are continuous, is automatically guaranteed with beam elements because they are only joined at their nodal points. The choice of beam elements for the spindle modeling is widely supported as demonstrated by Optiz and Noppen [45,p.228] who state, "For structures of the type of a spindle the mathematical model represents the true state of stress very well and therefore a beam approximation is sufficient."

The Beammod subroutine models the spindle-bearing configuration based on beam type elements and the description stored in file BEAM.DAT. The cross section properties of the spindle sections are calculated from the internal and external dimensions of the corresponding element. Additional data concerning material properties, constraints, and concentrated masses and flexibilities are used. The beam elements are used founded on the assumptions that

- the overall elastic behavior is not significantly influenced by local deformations around applied loading points;
- the dimensions in one unique direction of the structure are superior to all others;
- no elastic deflection of cross sections takes place;
- plane sections remain plane.

The beam element model is organized to minimize memory storage and maintain comparatively small computing requirements for application on personal computers. Figure 5.14 illustrates the automated modeling of an example spindle. Beam elements are defined by contour changes of the spindle core and are depicted by the element numbers within circles. The DOF associated with the ends of the beam elements are numbered. The odd and even numbers represent translation and rotation, respectively. The spindle housing is assumed to be rigid and the bearings are modeled by springs.

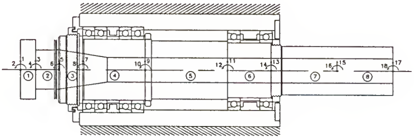


Figure 5.14 Beam Model of Spindle

5.10.2 Stiffness Matrix

The element stiffness matrices (K_i^e), corresponding to the global DOF of the structural model are calculated and the total stiffness matrix (K), is formed by the addition of the element stiffness matrices such that

$$K = \sum_{i=1}^{NE} K_i^e \, .$$

The stiffness properties of a beam element are physically the end-forces that correspond to unit element end-displacements. These forces can be evaluated by solving the differential equations of equilibrium of the element when it is subjected to the appropriate boundary conditions. Since by virtue of the solution of the differential equations of equilibrium, the exact internal displacement and stiffness matrices are calculated. The solution fulfills stress equilibrium, compatibility, and constitutive (ie. completeness of representation) requirements [43].

The stiffness matrix for a beam element with uniform cross-sectional and material properties may be derived by applying the principle of virtual work and solving for each of the coefficients, k_{ij}. The principle of virtual work states that for an elastic system in equilibrium, the work

done by the external forces is equal to the work of the internal forces during virtual displacement, [2,47]. The external work (W_e), and the internal work (W_i), may be calculated by the following equations,

$$W_e = k_{ij}\ \delta_i$$

$$W_i = \int_0^L M(x)\ d\theta$$

where

k_{ij} = the element stiffness matrix coefficient at row i and column j

δ_i = linear and angular displacements at the nodal coordinates of the element (i=1-4)

$M(x)$ = bending moment of element as a function of the distance from the beam end

$d\theta$ = relative angular displacement of the section.

The resulting beam stiffness matrix is a 4x4 symmetric matrix defined in terms of the displacement and rotation coordinates at each end of the beam. The static deflection curves of a beam element are depicted in Figure 5.15. The differential equation for small transverse displacement of a beam is represented by the following,

$$EI \frac{d^2y}{dx^2} = M(x) \ .$$

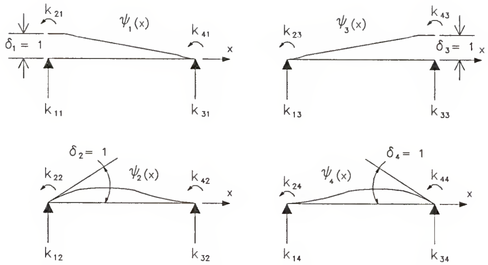


Figure 5.15 Static Deflection Curves for Beam Element

The general definition of the stiffness coefficients is designated by k_{ij} which corresponds to the force at nodal coordinate i due to a unit displacement at nodal coordinate j, maintaining all other nodal displacements at zero. The derivation of the element stiffness matrix coefficients will be demonstrated using $\psi_i(x)$. The beam deflection is subjected to the following boundary conditions,

$$x=0 \qquad y(0)=1 \qquad \frac{dy(0)}{dx} = 0$$

$$x=L \qquad y(L)=0 \qquad \frac{dy(L)}{dx} = 0 \ .$$

The moment is described by,

$$M(x) = k_{11}x - k_{21} \ .$$

Substituting for M(x) and then integrating twice yields

$$EI \frac{d^2y}{dx^2} = k_{11}x - k_{21}$$

$$EI \frac{dy}{dx} = \frac{k_{11}}{2} x^2 - k_{21}x + C_1$$

$$EI\ y = \frac{k_{11}}{6} x^3 - \frac{k_{21}}{2} x^2 + C_1x + C_2$$

The equations introduce the stiffness coefficients and constants of integration as unknowns. From the above equations together with the expressions of boundary conditions, the unknown values may be solved for yielding

$$C_1 = 0\ ,\quad C_2 = EI\ ,\quad k_{11} = EI/L^3\ ,\ \text{and}\quad k_{21} = 6EI/L^2\ .$$

Substituting the values into the last equation results in the expression of the deflection curve for the beam segment,

$$\psi_1(x) = 1 - 3(x/L)^2 + 2(x/L)^3\ .$$

In an analogous fashion the equations for the remaining deflection curves may be derived and expressed by

$$\psi_2(x) = x(1 - x/L)^2$$

$$\psi_3(x) = 3(x/L)^2 - 2(x/L)^3$$

$$\psi_4(x) = x^2(x/L - 1)/L\ .$$

The total deflection $y(x)$ at coordinate x due to arbitrary displacements at the nodal coordinates of the beam segment is given by superposition as

$$y(x) = \psi_1(x)\delta_1 + \psi_2(x)\delta_2 + \psi_3(x)\delta_3 + \psi_4(x)\delta_4 \ .$$

In general, any stiffness coefficient associated with beam flexure can be expressed as

$$k_{ij} = \int_0^L EI \ \psi_i''(x) \ \psi_j''(x) \ dx \ .$$

The equivalence $k_{ij} = k_{ji}$ is a particular case of Betti's Theorem, known as Maxwell's Reciprocal Theorem, [43], and results in a symmetric element stiffness matrix. The force vector F, may be equated to the nodal displacement vector by the element stiffness matrix as follows,

$$F = k^e \ \delta$$

where

$$k^e = \frac{E\ I}{L^3} \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \ .$$

The global stiffness matrix is assembled by the direct stiffness method. Any stiffness coefficient k_{ij} of the system

may be obtained by adding together the corresponding stiffness coefficients associated with those nodal coordinates.

5.10.3 Mass Matrix

The inertial effects of the beam models may be approximated by the lumped mass method or the consistent mass method. The simplest method for considering the inertial properties for a dynamic system is the lumped mass method. The mass of the structure is assigned to point masses which are placed at the nodal coordinates where translational displacements are defined (i.e. odd numbered DOF). Assuming a uniform beam segment of length L and a constant mass distribution per unit length of μ, lumped masses of $\mu L/2$ are applied at the beam nodal coordinates 1 and 3. The advantages of the lumped mass method is that less computational effort is required in forming the system mass matrix, the matrix is diagonal, and it permits the elimination of rotational DOF by static condensation. Unfortunately the method requires the spindle be modeled by more beam elements for the course model does not sufficiently represent the proper mass distribution and therefore yields less accurate analysis results.

The consistent mass method assures a mass distribution consistent with the static deflection of the beam because the same interpolation functions are employed in the calculation of the load vectors and the mass matrix as in the evaluation

of the stiffness matrix. In the consistent mass method, it is assumed that the deflections resulting from a unit dynamic displacement or unit acceleration, at the nodal coordinates of the beam element may be described by the same functions, ψ_{1-4}, which were obtained from static considerations. In an analogous fashion, the external virtual work may be represented by, $W_e = m_{12}\delta_1$, where $\delta_1=1$. The virtual work due to internal forces (f_i) may be expressed by

$$dW_i = f_i(x)\psi_1(x) \qquad \text{where} \quad f_i = \mu(x)\ddot{y}_2(x) \ ,$$

therefore yielding

$$dW_i = \mu(x)\psi_2(x)\psi_1(x) \ .$$

The internal virtual work for the entire beam segment must be integrated over the length,

$$W_i = \int_0^L \mu(x)\psi_2(x)\psi_1(x)\,dx$$

In general, a consistent mass coefficient may be calculated by equating the external and internal virtual work,

$$m_{ij} = \int_0^L \mu(x)\psi_i(x)\psi_j(x)\,dx \ .$$

The consistent element mass matrix of the beam segment relates the inertial force vector to the unit accelerations which are applied at the nodal coordinates $F = m^e\delta$, where

$$m^e = \frac{\mu L}{420}\begin{bmatrix} 156 & 22L & 54 & -13L \\ 22L & 4L^2 & 13L & -3L^2 \\ 54 & 13L & 156 & -22L \\ -13L & -3L^2 & -22L & 4L^2 \end{bmatrix} .$$

The complete system mass matrix is assembled in an identical manner as the stiffness matrix. The primary output of the Spindle Modeling module are the real symmetric stiffness and mass matrices modeling the spindle-bearing system and used by the Analysis module. The user may also direct the stiffness and mass output to be written to a file in column form, where the system size is followed by the columns of the stiffness matrix and then the mass matrix columns.

5.11 Spindle Analysis Module

The Spindle Analysis module performs the dynamic analysis (eigensolution) on the spindle model defined by the Spindle Modeling module. The analysis yields the natural frequencies and mode shapes of the spindle, as well as the modal

stiffnesses and masses normalized to the spindle tool tip. The module consists of 1 main subroutine and 7 support subroutines. A variety of eigensolution methods were written and tested in order to determine an efficient and accurate method to calculate the natural frequencies and mode shapes. The number of modes of interest and frequency range varies based on the characteristics and parameters of the machining operations.

Four eigensolution methods were investigated. The main criteria by which the methods were judged was the speed and accuracy in which the eigenvalues and eigenvectors were calculated. The Inverse Power Method with a dynamical matrix defined as $D=K^{-1}M$ does have two main advantages; the eigenvalues are found in ascending order, and the eigenvectors do not have to be back-transformed. The dynamical matrix was not symmetric in general. Unfortunately the accuracy of the previously calculated eigenvectors greatly influenced both the eigenvalues and eigenvectors in subsequent calculations. This is due to the fact that each eigenvector is used to form a sweeping matrix by which the dynamical matrix is multiplied (resulting in a deflation). Finally, the Inverse Power Method required many iterations to converge sufficiently, therefore the method was prohibitively slow for the analysis of spindles. Convergence was measured by the change in the eigenvector norm.

A modified Determinant Search Method proved to be fairly successful in finding the eigenvalues of system matrices of

nearly any form. The sign of the determinant was found by using Gaussian elimination to form an upper-triangular system matrix and then recording the number of sign changes on the main diagonal. By monitoring the sign of the determinant, the intervals containing roots to the characteristic equation could be repeatedly narrowed until the eigenvalue was accurately determined. The method required approximately $12N^3$ floating-point operations (FLOPS) to find all of the system eigenvalues. The FLOP estimates are based on modified calculations in Golub [24]. The method was determined to be too slow for Personal Computer applications and some complications, occasionally resulting in a missed eigenvalue, were caused by cases with closely spaced roots.

EISPACK and IMSL routines generally view root searching in the characteristic equation as an ineffective method for finding eigenvalues because the characteristic polynomial is of order N [49]. Most of these routines are based on Sturm's theorem discussed in [43]. A sequence of polynomials of lower degree produced by a recurrent form, a Sturm sequence, is used to localize eigenvalues to intervals on the real axis. Roots over the interval are found by the Bisection Method, which is occasionally enhanced by use of Newton's Method also. The corresponding eigenvalues are found by an inverse iteration. Finding all of the eigenvalues of a system matrix requires approximately $6N^3$ FLOPS usually in double precision. This FLOP count also includes the computations required to tridiagonalize the symmetric dynamical matrix.

Eigens is the principal subroutine of the Spindle Analysis module. Eigens solves for the eigenvalues and eigenvectors of an NxN system. A symmetric dynamical matrix is formed using the method of Cholesky decomposition of the mass matrix by subroutine Cholesky,

$$D = M^{-\frac{1}{2}T} \; K \; M^{-\frac{1}{2}}$$

where

D = symmetric system dynamical matrix (positive definite)

M = spindle mass matrix

K = spindle stiffness .

The classical eigenproblem is established by the equation,

$$D \; X = \lambda \; X$$

where

λ = eigenvalue solution (w_n^2),

X = matrix whose columns are the eigenvectors corresponding to each eigenvalue,

w_n = natural frequencies of the spindle-bearing system.

The dynamical matrix is tridiagonalized using Householder reflections by subroutine Tred. The routine reduces an NxN symmetric matrix into a tridiagonal form by N-2 orthogonal transformations. Each transformation annihilates the required part of the column and corresponding row. The technique was

chosen over the more common Given's Method because it is more efficient, requiring half as many multiplications. The eigenvalues of the tridiagonal matrix are found by a modified QL algorithm with implicit shifts. Since the rate of convergence is proportional to the ratio of successive eigenvalues, convergence can be greatly accelerated, especially where eigenvalues are close, by the technique of shifting. The eigenvalues are found by subroutine TQL which finds all of the system roots but does not necessarily find them in a specific order, therefore the solution is heap sorted into ascending order by subroutine SORT. A heap sort is used because it is a true in-place sorting routine which requires no additional storage and a maximum of $N\log_2 N$ manipulations. The calculation of all of the eigenvalues requires approximately $2N^3$ FLOPS. The sorted vector is scanned to determine the number of eigenvalues which fall within the range of interest. The eigenvectors are calculated by

$$\mathrm{Vec} = A^{-1}\ B$$

where

Vec = the eigenvector corresponding λ

$A = [\ K - \lambda\ M]$

B = a "zero" vector with 1 in the normalizing row (i.e. {1,0,0,...,0}) .

The advantage of finding eigenvectors by this method over orthogonal transformations, sweeping methods, or deflation techniques is that each eigenvector is calculated independently. In addition, transformation matrices do not need to be calculated or stored. The accuracy of the eigenvector does not depend on prior calculations of eigenvectors or the order in which the vectors are determined. Sufficiently accurate results may be found with single precision operations and storage thus increasing the computational speed.

The accuracy of the calculated eigenvalues and eigenvectors from Eigens were compared to both published problems and the finite element program COSMOS/M. The module produced accurate solutions which were additionally used to calculate modal mass and modal stiffness values. The modal values were also in agreement with other sources.

5.12 Modal Graphics Module

Subroutine Gmodes comprises the Modal Graphics module and is responsible for the graphical display of the characteristic mode shapes of the spindle. The geometry of the design is read from file BEAM.DAT and is automatically scaled to account for the chosen unit system, monitor resolution, and aspect ratio. The spindle is collapsed down to its center line and the location of the bearings are depicted as shown in Figure

5.16. The spindle and tool are oriented such that the work-end of the spindle is on the left side of the screen. The mode shapes corresponding to each of the natural frequencies within the specified range are plotted and the modal parameters are reported at the top left corner of the screen. The mode shape is the dashed line and on color monitors has a different color than the center line. In addition, a color monitor depicts the separation of model elements by a single pixel on the center line which is the same color as the mode shape. From the graphical display of the spindle mode shapes the designer may interpret and understand possible sources of spindle flexibilities.

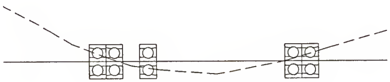


Figure 5.16 Depiction of Spindle Mode Shape

5.13 Graphics Transfer Function Module

The Graphic Transfer Function module, Gmodetf, calculates and displays the direct transfer function G, representing the

ratio of vibration X, over the radial or normal component of the force acting on the end of the tool, F_x. The transfer functions are covered in detail by Tlusty [69]. Because of the assumptions made in the Machine module, specifically a single degree-of-freedom process model and rigid workpiece, the direct transfer function is expressed as the modal superposition of the frequency response ratios of the individual modes of vibration of the spindle. The response for the n'th mode of vibration is expressed by

$$G_n = \frac{X(w)}{F_x(w)} = \frac{(1/k)\ w_n^{\,2}}{w_n^{\,2} - w^2 + 2j\zeta w_n w}$$

where

w_n = the natural frequency (rad/s)

ζ = the damping ratio = c/c_{cr} = $c/(2\sqrt{km})$

(1/k) = the static flexibility of the mode.

The real and imaginary parts of the direct transfer function are plotted on the screen and are calculated for the j modes of interest by the following,

$$\mathrm{Re}[G] = \sum_{n=1}^{j} \mathrm{Re}[G_n] \quad \text{and} \quad \mathrm{Im}[G] = \sum_{n=1}^{j} \mathrm{Im}[G_n]$$

where

$$\mathrm{Re}[G_n] = \frac{1}{k}\ \frac{1-(w/w_n)^2}{[1-(w/w_n)^2]^2 + (2\zeta w/w_n)^2}$$

$$\mathrm{Im}[G_n] = \frac{1}{k} \frac{2\zeta(w/w_n)^2}{[1-(w/w_n)^2]^2 + (2\zeta w/w_n)^2} .$$

The user may specify separate damping ratios for each of the modes or accept a default ratio of .03 (3%). For convenience and quick comparison to test data, the transfer function may be plotted as Displacement/Force, Velocity/Force, or Acceleration/Force. In addition, the calculated values may be written to the output file TF.OUT. The axes are automatically scaled to provide a clear representation of the real and imaginary plots.

5.14 Diagnostic Module

Subroutine Diagnos is used to determine the mode of vibration within the range of interest which critically limits the stability of the machining operation and diagnoses the primary cause of flexibility. The module informs the user if the present conceptual design meets the objectives as described by the design parameters. The user may choose to have the system explain how the critical mode is determined, how the cause of flexibility is determined, or which corrective actions are suggested. If the analysis of the mode shapes does not clearly suggest a cause of flexibility, the program returns an UNDETERMINED diagnosis, and informs the user how to inspect the transfer function and mode shapes.

The designer may then indicated to the system his opinion within the options provided of the principle cause of spindle flexibility. Corrective alternatives may be explored interactively by the designer, the design and analysis may be terminated, or considered complete. The interactive nature of the design system allows the novice designer to rapidly explore and understand the effects of design modifications. The corrective actions are listed in an order suggesting the appropriate sequence of operations to be explored.

A brief example of the Diagnos subroutine and the suggestion of corrective actions is listed below. The diagnosis below is based on the analysis of a conceptual model of the Setco test spindle with a 100 mm long, 25 mm diameter, endmill used as the tool.

```
Program Output:

The primary cause of flexibility for the critical
mode is the  TOOL BENDING.


          OPTIONS
 --------------------------------
  0   No action, EXIT module
  1   Critical mode explanation
  2   Diagnostic explanation
  3   Corrective actions

 Enter Option Number : 3


       Corrective Actions
 ---------------------------------
  0   No corrective action
  1   Eliminate tool extensions if used
  2   Reduce tool length
  3   Use integral tool/tool holder combination

 Enter Corrective Action : 0
```

5.15 Report Module

The Report module is used to summarize the final conceptual design of the specified spindle. The configuration, bearing information, and geometry is listed in an easily read format. This module may be further expanded to write script files or custom additions for the input of geometry to Computer-Aided Design systems or programs. The report is written to file SPINDLE.OUT. The general format of the described preliminary spindle design listed:

Project Title

Configuration Description

Bearing Description

Suggested Lubrication Method and Seal Type

Tabular Geometry Description .

CHAPTER 6

EXAMPLES

During the development of the Spindle Design System, a bottom-up integration of software modules was used in order to develop the modules independently and maintain control flexibility. This modularity enables the user to access the system in a variety of fashions depending on the level of knowledge and design detail defined. The following chapter will illustrate the different modes in which the system may be accessed and the program functionality as well as describe the results of the examples.

6.1 Automated Design

In order to establish a preliminary spindle-bearing design from initial milling application descriptions, the user selects the automated design option. The example demonstrates the description of a single purpose spindle. From the user's application description the performance requirements of the spindle design are determined. In this way both the form and function of the spindle may be represented in the design.

The interactive nature of the system is depicted and the results from the design session are compared to a commercially available milling spindle manufactured by Setco. Additional attention and comments which are not part of the system output with be enclosed in a box appearing to the right of the program dialogue. The user's responses appear to the right of the system prompt (:), or directly following a question.

Program Output:

```
  CODE           OPTION
------------------------------
    0      System Information
    1      Automated Design
    2      Bearing Stiffness
    3      Model Design
    4      Analysis

Enter Option Code : 0

 CODE          OPTION
----------------------------
   1     System Description
   2     Instructions

Enter Option Code : 2


This program is an interactive system which allows the
user to investigate why (?) a question is asked, or
request Help (H) in replying to system prompts.

Simply follow the interactive system by entering
option codes, requested values, ?, or H.

To permanently exit the system prematurely, press
Ctrl and C (ie Ctrl-C) simultaneously at any system
prompt.
```

```
   CODE            OPTION
------------------------------
    0      System Information
    1      Automated Design
    2      Model Design
    3      Analysis

Enter Option Code : 1


   Code         Units
--------------------------
   0     English (lb,in.
   1     Metric  (N,m

Enter Unit Code : 0
```

The user may ask for (H)elp in response to any system prompt.

```
Do you want to design a (S)ingle purpose
or (M)ulti-purpose spindle (S/M)? HELP
```

If the spindle is being designed to perform one specific machining operation or group of very similar operations, such as a spindle used in a transfer line, a (S)ingle purpose design may be used. If the machining operations or cutting conditions vary, as in spindles designed for machining centers, a (M)ulti-purpose spindle design is necessary.

```
Do you want to design a (S)ingle purpose
or (M)ulti-purpose spindle (S/M)? S


ENTERING WORKPIECE MATERIAL MODULE
```

The user is informed which module is being invoked.

```
Would you like to specify the workpiece Material,
Unit Power, and Cutting Stiffness directly (Y/N)?  ?
```

Why?

If you have sufficient experience with a particular workpiece material you may enter the Material name, Unit Power (Hp), and Cutting Stiffness (lb/in^2) directly, otherwise answer N to proceed.

```
Would you like to specify the workpiece Material,
Unit Power, and Cutting Stiffness directly (Y/N)?  N
```

```
 Code     Material
--------------------------
  1     Aluminum Alloys
  2     Cast Irons
  3     Copper Alloys
  4     High Temp Alloys (Ni, Co, Fe)
  5     Steels (Plain C, alloys, tool)
  6     Stainless Steel
  7     Titanium

Enter material code : 2


The hardness range for Cast Iron is 110-320 Bhn
Please enter workpiece hardness : ?  ?


The material hardness must be known in order
to determine Unit Power requirements, Cutting
Stiffness, and machining speeds and feeds

If the hardness is not known, choose a mid-
range value.

The hardness range for Cast Iron is 110-320 Bhn
Please enter workpiece hardness : 200


 Code     Tool Edge
------------------------
  1     High Speed Steel
  2     Uncoated Carbide
  3     Coated Carbide

Enter Tool Edge Code : ?


The tool material must be known to determine
allowable cutting speeds.


Would you like more explanation (Y/N)  : Y

Enter tool code for explanation : 1


High Speed Steel designates a number of steels
that have enhanced material properties.  High
Speed Steel can retain sufficient hardness at
temperatures up to 540-590°C (1000-1100°F), to enable
higher cutting speeds while the tool maintains
its original hardness. The principle alloying
elements are W, Mo, Cr, V, together with C.
```

```
 Code      Tool Edge
------------------------
   1     High Speed Steel
   2     Uncoated Carbide
   3     Coated Carbide

Enter Tool Edge Code : 1
```

The process is described by the definition of the desired milling parameters.

```
ENTERING MACHINING MODULE

 Code    Machining Operation
----------------------------------
   1        Face Milling
   2        End Milling

Enter machining operation code : 1

Enter the cutter diameter (in.) : 4

Enter the # of teeth, inserts, or flutes
on the cutter : 6

Enter the facemill thickness (in.) : 2.5

Enter maximum spindle speed (RPM) : 2000

The default chip load is .012 IPT,
do you want to reset max chip load (Y/N)? :  N

The default axial depth of cut is .250 in.,
do you want to reset depth of cut (Y/N)? : ?

The cutting parameters which influence cutting
stability and chatter vibrations may be expressed
as the Cumulative Chip Width, B.

              B = n b a/d
where
           n = number of teeth on cutter
           b = axial depth of cut
         a/d= radial width of cut over cutter dia.
              (often called the radial immersion).

Cutting parameters such as chip load and cutting
speed have little influence on the limit of stability
but they do affect the amplitude of vibration during
chatter.

The required stiffness of the spindle and tool,
assuming the workpiece rigid, is calculated using the
cumulative chip width and the cutting stiffness of
the workpiece. A slotting cut is assumed.
```

```
The default axial depth of cut is .250 in.,
do you want to reset depth of cut (Y/N)? : Y

Enter max depth of cut (in.) : 0.08


    DESIGN PARAMETER SUMMARY   1
-------------------------------------

WORKPIECE MATERIAL  : Cast Iron
MACHINING OPERATION : Face Milling
TOOL EDGE MATERIAL  : High Speed Steel
DIAMETER OF CUTTER  :     4.00 in.
NUMBER OF CUTTER TEETH :  6
MAX SPINDLE SPEED   :     2000. rpm
MAX CHIP LOAD       :     .012 ipt
MAX DEPTH OF CUT    :     .080 in.

AXIS FEED RATE REQ. :      144.0 in/min
METAL REMOVAL RATE  :        46. in^3/min
CUTTING SPEED       :      2094. sfpm
SPINDLE POWER REQ.  :       40.6 Hp
SPINDLE STIFFNESS REQ. : 962743. lb/in
TOOTH IMPACT FREQ.  :       200. Hz
```

A review of the performance requirements from which the spindle is designed.

```
ENTERING SPINDLE GEOMETRY MODULE


Enter spindle nose size (30,40,45,50) : 60


*** invalid response ***
```

User responses are checked for their validity (ex. a specified I.D. cannot be greater than the O.D.)

```
Enter spindle nose size (30,40,45,50) : H

The spindle nose size is used to determine
the taper holding the tool shank.

The following table indicates the nose size
and corresponding gage diameter of taper.
```

ANSI Standard

Nose Size	Gage Dia.(in.)	Gage Dia.(mm)
30	1.25	31.75
40	1.75	44.45
45	2.25	57.15
50	2.75	69.85

Enter spindle nose size (30,40,45,50) : 50

Constrain spindle length (Y/N)? : H

If a minimum or maximum spindle length is required then answer Y. Design constraints corresponding to length specifications will be imposed.

Constrain spindle length (Y/N)? : N

User specified design constraints.

Constrain spindle overhang (Y/N)? : N

Design for tool draw bar (Y/N)? : Y

Design for tool extension (Y/N)? : ?

If a tool extension will be included in the primary use of the spindle, the analysis of the dynamic stiffness of the spindle and tool will include the dynamic behavior of a tool extension as defined by its geometry.

The primary use of a tool extension is to increase spindle reach in areas of little clearance. It should be noted that a tool extension will result in additional spindle flexibility and therefore will increase chances of instability during machining.

Design for tool extension (Y/N)? : N

ENTERING BEARING MODULE

Will the spindle be mounted vertically (Y/N) ? N

Spindle configuration 5 has been selected with GREASE lubrication and LABYRINTH seals.

The estimated front bearing set stiffness is 8.22E+06 lb/in.

The estimated rear bearing set stiffness is 2.39E+06 lb/in.

ENTERING INITIAL OPTIMIZATION MODULE

```
# ITERATION =          82
OVERHANG    :        3.00 in.
BEARING SPAN:       14.00 in.
STIFFNESS   : 1699463.00 lb/in
```

Design optimized for min. static flexibility at the tool. The stiffness is initially predicted to ensure design feasibility.

```
ENTERING PRELIMINARY SPINDLE GRAPHICS MODULE

  FIRST APPROXIMATION OF OPTIMIZED SPINDLE


ENTERING PRELIMINARY SPINDLE MODELING MODULE


ENTERING SPINDLE MODELING MODULE


ENTERING SPINDLE ANALYSIS MODULE

Continue to Analysis (Y/N) ?
```

The user may stop and analyze the preliminary spindle design later.

```
Solving for Eigenvalues...


Eigenvalue           1 =    3687.177000 rad/s
Eigenvalue           2 =    4785.935000 rad/s
Eigenvalue           3 =    8252.940000 rad/s
Eigenvalue           4 =   15421.460000 rad/s
Eigenvalue           5 =   23080.960000 rad/s
Eigenvalue           6 =   36334.440000 rad/s
Eigenvalue           7 =   53612.630000 rad/s
Eigenvalue           8 =   82284.680000 rad/s
Eigenvalue           9 =  120778.400000 rad/s
Eigenvalue          10 =  162821.200000 rad/s
Eigenvalue          11 =  202999.000000 rad/s
Eigenvalue          12 =  255534.500000 rad/s
Eigenvalue          13 =  362314.400000 rad/s
Eigenvalue          14 =  464738.500000 rad/s
Eigenvalue          15 =  546269.900000 rad/s
Eigenvalue          16 =  736785.800000 rad/s
Eigenvalue          17 = 1317657.000000 rad/s
Eigenvalue          18 = 2019120.000000 rad/s

NUMBER OF MODES OF INTEREST =           1

Solving for Eigenvectors...

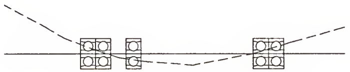
Would you like to write Eigenvalues and Mode Shapes
to file EIGENS.OUT  (Y/N)? N


ENTERING MODAL GRAPHICS MODULE
```

MODE 1 = 3687.177 RAD/S

MODAL STIFFNESS = .373892E+06 lb/in
MODAL MASS = .275016E-01 lb

Units always follow output. Mass is in lbm.



Do you want to view modes again (Y/N)? N

ENTERING TRANSFER FUNCTION MODULE

The assumed damping ratio for each mode is .03 (3%)
Would you like to reset the damping ratio for each mode (Y/N)? N

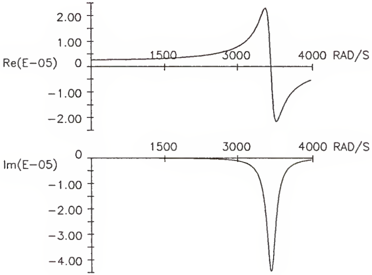
Code	Transfer Function Type
1	Displacement/Force
2	Velocity/Force
3	Acceleration/Force

Enter Transfer Function Code : 1

Would you like to write Transfer Function to file TF.OUT (Y/N)? Y

The transfer function may be calculated and plotted in the most convenient form. The screen plot may be captured by means of an EGA screen dump to an Epson compatible dot matrix printer. The following plot was obtained from the calculated transfer function data written to the file TF.OUT.

DISPLACEMENT/FORCE TRANSFER FUNCTION
DIRECT AT TOOL POINT FOR FIRST 1 MODES



Would you like to recalculate the T.F. (Y/N)?

ENTERING MODAL DIAGNOSTIC MODULE

DIAGNOSIS :

The critical design mode is mode 1

The primary cause of flexibility for the critical mode appears to be the first BENDING mode of the spindle.

The spindle behavior is adequate for application(s) specified.

OPTIONS
----
0 No action, EXIT module
1 Critical mode explanation
2 Diagnostic explanation
3 Corrective actions

Enter Option Number : 2

The preliminary design does not have to be modified to meet the design objectives.

Explanation of the diagnosis aids the user's understanding of the evaluation.

```
The primary cause of the flexibility of the critical
mode is determined by examining the spindle mode
shape corresponding to the critical mode.

By comparing the relative displacement values of
the mode shape at the tool end, bearing locations,
spindle span between bearings, and the drive end,
the primary source of the modal flexibility is
determined.

 Would you like to view the mode shape again (Y/N) ? N


          OPTIONS
--------------------------------
 0   No action, EXIT module
 1   Critical mode explanation
 2   Diagnostic explanation
 3   Corrective actions

Enter Option Number : 0


ENTERING REPORT MODULE


Stop - Program terminated.
```

The automated design process resulted in the specification of spindle-bearing configuration V as described in Section 4.6. The front bearing bore size of the 15° angular contact ball bearings is 3.15 in. (80 mm), and the rear bearings are 2.75 in. (corresponding to 70 mm), in diameter. Founded on the characterized speed of the spindle (ie. 135,000<DN<500,000), grease lubrication and Labyrinth seals are recommended. The limiting axial depth of cut based on the dynamic stiffness of the preliminary spindle design is b_{lim}=0.115 in. Because the tooth impact frequency of 200 Hz is low relative the frequency of the critical first mode of

the spindle at 587 Hz (3687 rad/s), the real transfer function is always positive in the vicinity of the excitation frequency and therefore the process is not limited by chatter. The spindle-bearing configuration V is depicted in Figure 6.1 b). The geometry of the preliminary design is described by Table 6.1 in which the dimensions of the numbered spindle segments are defined. The housing shown for the configuration is for illustrative purposes only for the Spindle Design System does not specify the housing. The spindle-bearing configuration does however facilitate the stepped design of the inner housing allowing the housing to be machined from one direction. The figure compares the 4307M Setco milling spindle portrayed in Figure 6.1 a) to the preliminary spindle configuration resulting from the design session. The Setco spindle was sized based on the milling spindle selection chart in the catalog, [59]. An application which requires 41 horsepower at 2000 rpm implies the selection of an 07 spindle size. The catalog also specifies grease lubrication and Labyrinth seals for rotational speeds between 1700 rpm and 6200 rpm.

Table 6.1 Spindle Geometry

Segment No.	Length (in.)	OD (in.)	ID (in.)
1	3.00	5.00	-
2	2.80	3.15	-
3	4.20	3.15	1.10
4	7.00	2.95	1.10
5	2.75	2.75	1.10
6	5.50	2.55	1.10

A)

B)

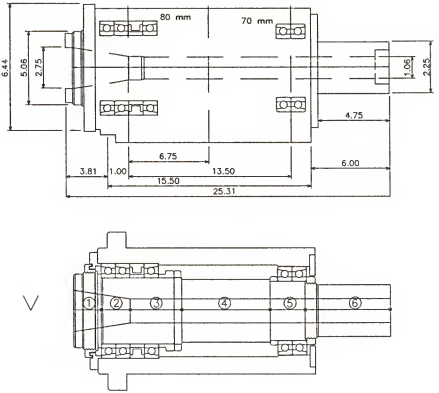


Figure 6.1 Spindle Design Comparison a) Setco Spindle b) Preliminary Spindle Design

6.2 Bearing Stiffness

The load-deflection characteristics for spindle bearings may be investigated independently from the spindle design process through the choice of the bearing stiffness option. The estimation of the bearing stiffness is made within the Bearings module and is dependent on the level of detailed knowledge of the user. A quick estimation of the radial stiffness is founded on empirical relations which are based on the bearing bore diameter and preload. This approximation

is usually sufficient during the automated design process because the detailed geometry information required for a more complete analysis is not known until a specific bearing has been selected. The user may however investigate the behavior of radial bearings and angular contact bearings in detail through the utilization of subroutines Radial and Combined. These subroutines are described and demonstrated in Sections 5.6.1 and 5.6.2, respectively.

In this example the effect of preload on a spindle-bearing configuration will be described with respect to the method of application of the preload. As mentioned before, angular contact ball bearings and taper roller bearings are preloaded by the axial displacement of one of the races relative to the other. The preload results in a uniform loading of the rolling elements against both races which is proportional to the displacement and the contact angle. Angular contact bearings are usually employed in pairs in order to support a combination of radial and thrust loads. Because the bearing pairs are preloaded against one another, an overall force balance for the spindle-bearing configuration is required to accurately calculate the load-deflection behavior of a specific bearing. Two preloading methods are depicted in Figure 6.2. The bearings in the example are Fafnir 9120 angular contact ball bearings with an inner race diameter, D_i=109.5 mm, an outer race diameter, D_0=140.5 mm, yielding a pitch diameter of 125 mm. The ball diameter is d=15.5 mm, and the radius of the race groove is r=8.06 mm.

The resulting conformity ratio between the race and the ball is 0.52. The bearings have a free contact angle (β_f), of 15° with 22 rolling elements in a single row.

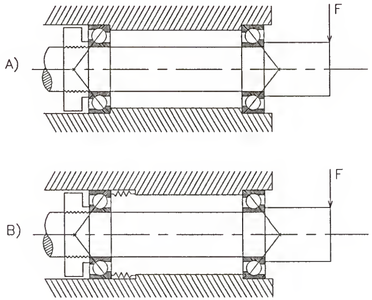


Figure 6.2 Bearing Preload Methods

In case A), the pair of bearings are preloaded by a constant distance method which results in a variable preload. The preload is installed during the installation of the spindle by means of an adjusting nut which displaces the inner races of the bearings by a constant distance. Both of the bearings are preloaded by an axial displacement (δ_{ap}), equal in magnitude and opposite in direction, which consistent with the direction of the azimuth angle for each of the bearings. The preload method provides no mechanism to maintain the preloading force on the bearings. Because the bearings oppose

each other, the axial displacement of the spindle shaft increases the load on one bearing while relieving the load on the other. In addition, the radial force F, is proportionally supported by the rear bearing (F_r = F(A/B)), and the front bearing (F_f=F(A+B)/B), resulting in varied contact angles of the bearings during loading. The bearing span to overhang ratio (B/A), used in the examples is 2. Such a preloading method is rarely employed for long spindles because the thermal expansion of the spindle shaft significantly reduces the preload of both of the bearings. The load-deflection behavior of the front bearing for the constant distance preload method is illustrated in Figure 6.3. The load-deflection coefficient of the Fafnir bearing was calculated as 3.98×10^5 $N/mm_{1.5}$. An axial preload for the bearings was applied, δ_{ap}=0.02 mm, corresponding to a normal deflection of the balls of 0.0052 mm and a normal force on each element of 148 N. The axial force (F_{ap}) associated with preload was 1024 N. The contact angle of the bearing increased as a result of the applied preload to β=18.3°. As shown in Figure 6.3, the radial stiffness of the front bearing is nearly linear as the applied force on the spindle initially increases. The bearing exhibits a radial stiffness of 4.2×10^5 N/mm until the preload is relieved at F=1800 N. At this point the normal load on the top element goes to zero and the ball may lose contact with the outer race. The contact angle has also decreased to 17.9° in order to support the applied radial load. Because the

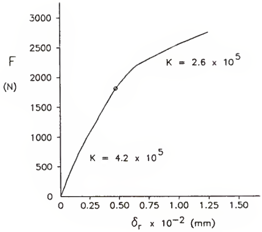


Figure 6.3 Case A, Constant Distance

applied load is distributed over fewer rolling elements, the stiffness of the bearing is significantly reduced. The bearing stiffness in this case decreases by 38% to a stiffness of 2.6×10^5 N/mm. This loss of stiffness and the cyclical loading of the rolling elements would result in premature failure of the bearing during spindle operation.

In case B) however, the rear bearing is not located by the shoulder of the housing and is therefore free to float axially. The amount of preload on the bearings may be modified by the use of a hydraulic cylinder represented in Figure 6.2 by the spring between the housing an the rear bearing. In case B) a constant preloading force is maintained equally on both bearings, therefore the axial force acting on each bearing is known at all times. The differing axial deflections are due to the proportional support of the

external radial load. If the sum of the axial components at one bearing exceeds the opposing external axial load, the bearing will move in the direction of the axial component of the radial load. The load-deflection behavior of the front bearing under the constant preload force of case B) is illustrated below in Figure 6.4.

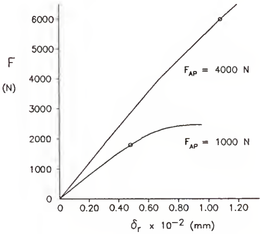


Figure 6.4 Case B, Constant Force

The figure illustrates the significant influence that the preload has on the exhibited radial stiffness. The bottom curve corresponds to the constant axial preload of 1000 N corresponding to δ_{ap}=0.0197 mm. The preload force is distributed equally (145 N/ball), on the rolling elements and the contact angle is increased to 18.2°. The stiffness of the front bearing preloaded with F_{ap}=1000 N is 4.2x10^5 N/mm until the axial force component on the bearing causes the translation of the rear bearing. At this point the stiffness

exhibited by the front bearing drops off rapidly and the balancing of the overall forces becomes ill-conditioned. The upper curve represents a 49% increase in the radial stiffness of the bearing during to the constant preload of 4000 N. The corresponding axial preload is 0.0434 mm. The preload force is distributed as 493 N/ball and the initial contact angle due to the preload is 21.7°. The load-deflection behavior of the bearing is nearly linear over the external loading range and the exhibited stiffness is 6.1×10^5 N/mm. The characteristic loss of stiffness is not demonstrated until F=6000 N.

The example illustrates bearing stiffness analysis and the significant influence of preload on the bearing stiffness. The study of the cases presented were compared to the analysis of the spindle considering only the radial stiffness of the front and rear bearings. The simplified representation of the radial behavior of the bearings proves to be accurate provided that the externally applied radial forces which are typically encountered are proportionally distributed among the bearings prior to the estimation of the bearing stiffness and adequate preload is supplied to maintain the nearly linear behavior.

6.3 Model Design

The model design option of the system may be employed to interactively describe a model of an existing design or spindle. The objective of modeling the Setco test spindle for

the purpose of this example is to demonstrate that the analysis of a conceptual design may accurately predict the dynamic behavior of the final product. Although the simplified model of the spindle does not include the detailed geometry of the final design, the analysis can be used to evaluate the performance of the proposed spindle-bearing configuration and judge the relative frequency response behavior. The simplified Setco test spindle is depicted in Figure 6.5. The actual spindle contains a roller cage in which the rear bearing set is mounted. The cage permits the axial translation of the bearings although the small diameter of the numerous rollers distributed over the cage provides high radial stiffness compared to the rear bearings. The roller cage was therefore neglected in the description of the test spindle.

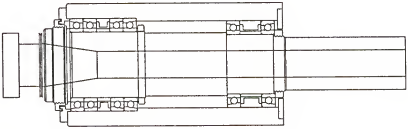


Figure 6.5 Setco Test Spindle

The test spindle behavior was measured by the direct transfer function at the tool using an accelerometer and an instrumented impact hammer. The test spindle is mounted to a large steel plate placed on mounting pads and was excited

horizontally to minimize the excitation of the flexible plate modes. The measured transfer function was curve fit yielding the modal parameters of the significant spindle modes measured below 1500 Hz. In Figure 6.6 the real and imaginary components of the direct transfer function are depicted for both the measured spindle and the simplified model. As indicated by the correspondence of the plots, the conceptual model of the spindle reasonably describes the dynamic behavior of the test spindle and indicates the relative importance of the first and second modes.

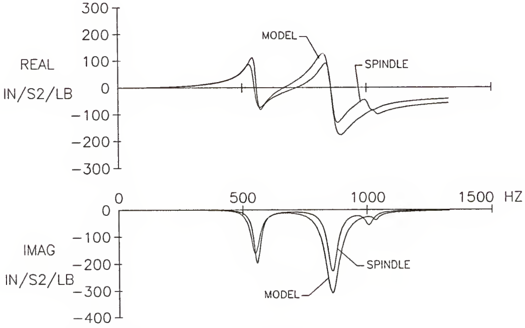


Figure 6.6 Transfer Function Comparison

Examination of the modeled mode shapes indicate that the first mode is primarily due to the overhang of the tool and front bearing deflection while the critical second mode exhibits the flexure of the spindle shaft with some bearing contributions.

Each of the examples presented have indicated the functionality of Spindle Design System and illustrated the ability to aid a novice designer in the design of a reasonable, well conceived spindle-bearing system utilizing existing spindle design knowledge and empirical data. In addition, the interactive nature of the system provides a means by which the influence of design parameters may be understood and the design alternatives readily evaluated.

CONCLUSIONS

The modern machine tool industry is undergoing rapid change and development. Global competition, increasing productivity demands, and the desire to exploit the potentials of machines through increased automation have resulted in greater requirements of machine tools and machine tool spindles. The development of a new generation of machine tool spindles is guided by market demands, new technology, and machine design. The performance objectives of higher speeds and greater power utilization combined with increased accuracy of the machined surface have magnified the importance of the spindle design process and accentuated the need to develop and integrate computer-based techniques in design.

Engineering design is a creative process that is essential to the development and refinement of new products. The very nature of real engineering, manufacturing, and design problems have challenged the implementation of computer-based aides. These problems are often ill-posed, open-ended, and lack well-defined objectives. Contemporary CAD/CAM systems are most useful for the revision and documentation of nearly completed designs. The CAD/CAM models are mainly employed as input to analysis routines and to drive CAM modules. In addition, the designer typically has little control or

understanding of the process parameters influencing the design other than general overall objectives. As a result, the critical tradeoffs, between information, cost, and the ability to change the design are often overlooked. In the earliest stages of the design process there is relatively little detailed product or performance information, although this lack of definition allows for the greatest design flexibility at the lowest cost. Previously CAD/CAM and expert system approaches required a relatively high level of product definition prior to the use of any automated methods. Consequently, the feedback resulting in design changes occurred late in the design process when modifications are made under more constraints and at an increased cost.

In the Spindle Design System, the engineering relationships governing the design of high-speed spindles are defined by analytically deterministic models of the milling process and spindle-bearing component behavior. Heuristics are additionally used to represent empirical data founded on speed, size, and lubrication information. "Rules of thumb" are employed to guide the selection of appropriate bearings and spindle-bearing configurations. In this way, the process information is integrated with the initial design phases of the spindle and realistic performance characteristics may be determined. The system provides a means by which the functional requirements of a spindle may be used to define performance objectives and design parameters to enhance the preliminary design process. The design system may also be

applied to aid in the more complete specification of spindle criteria including structural characteristics prior to the purchase of a commercial spindle.

The functional behavior of the Spindle Design System illustrates the integrated definition of the spindle based on form and function. The design and analysis activities are responsible for the definition of the spindle form. Typically CAD/CAM approaches to design only address the product by its physical form designated by material, shape, tolerances, and assembly specifications, while essentially neglecting the product function. In the knowledge-based system described, the function of the spindle is determined through application descriptions of machining processes and performance objectives constitute a driving mechanism to the design process.

The interactive design system places the user at the center of the conceptual design process by aiding in the definition and representation of function requirements as well as by providing a means of design evaluation. Throughout the design process the user maintains the role of decision-maker while the system provides updated information.

Founded on a knowledge-based approach, an "optimized compromise" to the often conflicting spindle design objectives is presented in the form of a conceptual design. Despite the knowledge gap which exists during initial design stages, sound decisions and evaluations of design alternatives are made. A reduction of trial and error iterations may be achieved by the early specification of the required spindle performance.

The underlying objectives and constraints governing the spindle design may be determined by the parameter definition of machining applications. Finally, a conceptual spindle design model is effectively utilized to predict the characteristic behavior of the final product despite the lack of detailed design information. A more complete evaluation of the conceptual design may significantly reduce the need for exhaustive verification of a spindle-bearing configuration later in the design process and allows for more timely feedback resulting in effective design modification.

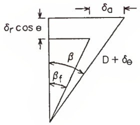
The knowledge-based system based on the hierarchical integration of design task modules demonstrates an effective design methodology which couples the array of disorganized knowledge in the domain of spindle design with well established numeric techniques. While aiding in the design and evaluation of high-speed spindles, the development of the interactive design system discloses the critical need for a greater understanding of the modes of spindle bearing failure and deterministic models which are not subject to the specific limitations of the test arrangements and procedures. In this way, high-speed spindle bearings may be specified deterministically rather than on the DN values.

Finally, knowledge-based systems may be used to represent and demonstrate existing design methodologies, although further development and modeling of the creative design process is required to effectively generate realistic and innovative design alternatives.

APPENDIX

DERIVATION OF THE NEWTON-RAPHSON EQUATIONS FOR ACBB

A summary of the complete derivative of the equations for the loaded contact angle as well as the error derivatives are shown. The derivatives for the axial error (A'), and the radial error (R'), are summarized by the following,



$$(D+\delta_\theta)^2 = (D\cos\beta_f + \delta_r\cos\theta)^2 + (D\sin\beta_f + \delta_a)^2$$

$$\delta_\theta = D\left[(D\cos\beta_f + \delta_r\cos\theta)^2 + (D\sin\beta_f + \delta_o)^2\right]^{1/2} - D$$

$$DEN = \left[(D\cos\beta_f + \delta_r\cos\theta)^2 + (D\sin\beta_f + \delta_a)^2\right]^{1/2}$$

$$\delta_\theta = D\,(DEN - 1)$$

$$F_\theta = K_n\,\delta_\theta^{\,q}$$

$$\cos\beta = (\cos\beta_f + \frac{\delta_r}{D}\cos\theta)/DEN$$

$$\sin\beta = (\sin\beta_f + \frac{\delta_a}{D})/DEN$$

$$F_a = \Sigma\, F_\theta \sin\beta$$

$$A = F_a - \Sigma\, F_\theta \sin\beta = F_a - \Sigma K_n D^q \left\{\left[(\cos\beta_f + \frac{\delta_r}{D}\cos\theta)^2 + (\sin\beta_f + \frac{\delta_a}{D})^2\right]^{1/2} - 1\right\}^q \sin\beta$$

$$A = F_a - \frac{\Sigma K_n D^q \left\{\left[(\cos\beta_f + \frac{\delta_r}{D}\cos\theta)^2 + (\sin\beta_f + \frac{\delta_a}{D})^2\right]^{1/2} - 1\right\}^q (\sin\beta_f + \frac{\delta_a}{D})}{\left[(\cos\beta_f + \frac{\delta_r}{D}\cos\theta)^2 + (\sin\beta_f + \frac{\delta_a}{D})^2\right]^{1/2}}$$

$$A' = \left\{(\Sigma \frac{F_\theta}{D} q (\sin\beta_f + \frac{\delta_a}{D})^2 + DEN\,\Sigma \frac{F_\theta}{D}) - \Sigma F_\theta (\sin\beta_f + \frac{\delta_a}{D})^2 (D\;DEN)\right\} / DEN^2$$

$$A' = A' + q F_\theta \sin^2\beta / \delta_\theta + F_\theta/(D\;DEN) - F_\theta \sin^2\beta/(D\;DEN)$$

$$A' = A' + q F_\theta \sin^2\beta / \delta_\theta + F_\theta \cos^2\beta/(D\;DEN)$$

$$\delta_a = \delta_a + \frac{A}{A'}$$

$$F_r = \Sigma F_\theta \cos\theta \cos\beta$$

$$R = F_r - \frac{\Sigma K_n D^q \left\{\left[(\cos\beta_f + \frac{\delta_r}{D}\cos\theta)^2 + (\sin\beta_f + \frac{\delta_a}{D})^2\right]^{1/2} - 1\right\}^q \cos\theta (\cos\beta_f + \frac{\delta_r}{D}\cos\theta)}{\left[(\cos\beta_f + \frac{\delta_r}{D}\cos\theta)^2 + (\sin\beta_f + \frac{\delta_a}{D})^2\right]^{1/2}}$$

$$R' = R' + q F_\theta \cos^2\beta \cos^2\theta / \delta_\theta + F_\theta \cos^2\theta \sin^2\beta/(D\;DEN)$$

$$\delta_r = \delta_r + \frac{R}{R'}$$

REFERENCES

1. Aldridge, J., J. Cerutti, W. Draisin, M. Steverwalt, "Expert Assistants for Design," Los Alamos National Laboratory, LA-UR-85-3970, submitted to the First International Conf. on Application of AI to Engineering Problems, South Hampton, U.K., April, 1986.

2. Bathe, Klaus-Jurgen, Finite Element Procedures in Engineering Analysis, Prentice-Hall, Inc., Englewood Cliffs, NJ, 1982.

3. Betsch, H., "How Fast Can Machine Tool Spindles Operate?" Ball Bearing Journal No. 214, SKF Limited, August, 1983, pp. 1-13.

4. Chen, J.L., "A Knowledge Based Expert System Approach to Optimum Structural Synthesis," Ph.D. dissertation, University of Florida, Gainesville, FL, 1987.

5. Chen, Shu-Hsing, "Ball Motion and Temperature Rise in High Speed Angular Contact Ball Bearings," Ph.D. dissertation, University of Florida, 1989.

6. Chryssolouris, G., and K. Wright, "Knowledge-Based Systems in Manufacturing," Annals of the CIRP, Vol. 35(2), 1986, pp. 437-440.

7. Clough, R.W., "The Finite Element in Plane Stress Analysis," Proceedings of the 2nd A.S.C.E. Conf. on Electronic Computation, Pittsburgh, PA., Sept., 1960, pp. 1-13.

8. Courant, R., "Variational Methods for the Solution of Problems of Equilibrium and Vibrations," Bulletin of the American Mathematical Society, Vol 49, Jan., 1943, pp. 1-23.

9. Davis, D.B., "Artificial Intelligence Enters the Mainstream," High Technology, July, 1986, pp.16-23.

10. Deitz, D., "The Power of Parametrics," Mechanical Engineering, Jan., 1989, pp. 58-64.

11. Denning, P.J., "The Science of Computing," American Scientist, Vol. 74, Jan-Feb, 1986, pp. 18-20.

12. Deutchman, A.D., W.J. Michels, C.E. Wilson, Machine Design, MacMillan Publishing Co., Inc., New York, 1975.

13. Devorak, Paul J., editor, "1987 Mechanical Drives Reference Issue," Machine Design, Oct. 15 issue, 1987.

14. Draisin, W., and E. Peter, "An Intelligent Interface for Design and Simulation," Los Alamos National Laboratory, LA-UR-3275, submitted to the First International Conf. on Expert Database Systems, April 1-4, Charleston, SC, 1986, pp. 1-11.

15. Dym, C.L., "Expert Systems: New Approaches to Computer-Aided Engineering," Engineering with Computers, Vol. 1(1), Springer-Verlag, NY, NY, 1985, pp. 9-25.

16. Elsaie, A, "Optimizing Structural Design," CAE, Oct, 1986, pp. 70-76.

17. El-Sayed, H.R., "Bearing Stiffness and the Optimum Design of Machine Tool Spindles," Machinery and Production Engineering, Vol. 125, 1975, pp.519-524.

18. Eschmann, Hasbergen, and Weigand, Ball and Roller Bearings, John Wiley & Sons, New York, 1985.

19. Field, M., "High-Speed Machining: An Update," American Machinist, Feb., 1983, pp. 88-92.

20. Fox, R.L., Optimization Methods for Engineering Design, Addison-Wesley Publishing Co., Reading, MA, 1971.

21. Freeman, H., and P.M. Lewis II, editors, Software Engineering, Academic Press, New York, 1980.

22. Giebner, E., "Bearing Arrangement Design for Machine Tool Spindles," Ball Bearing Journal, No. 216, Oct, 1983, pp. 1-13.

23. Gilmore, J.F., and K. Pulaski, "A Survey on Expert System Tools," IEEE Proceedings of the 2nd Conference on Artificial Intelligence Applications, Miami Beach, FL, Dec. 11-13, 1985, pp. 498-502.

24. Golub, G.H., and C.F. Van Loan, Matrix Computations, Johns Hopkins University Press, Baltimore, MD, 1983.

25. Hamrock, B.J., and D. Dowson, Ball Bearing Lubrication, John Wiley & Sons, New York, 1981.

26. Harris, T.A., Roller Bearing Analysis, John Wiley & Sons, Inc, New York, 1984.

27. Hatvany, J., "Available and Missing AI Tools," Annals of the CIRP, Vol. 35(2), 1986, pp. 433-435.

28. Hernadez, I.A., "Design of High Speed, High Power Spindles Based on Roller Bearings," Ph.D. dissertation, University of Florida, 1988.

29. Hernandez, I.A., and S.H. Chen, "Load Deflection Behavior of Spindle Bearing Systems," Internal Report, University of Florida, Gainesville, FL, April, 1986.

30. Holovacs, J., "Program Locates Optimum Values," Machine Design, March 6, 1986, pp. 144-146.

31. Jie, Z., and C. Kechen, "Structural Dynamic Optimal Design Using Modal Model, Proceedings of the 6th International Modal Analysis Conf., Feb 1-4, 1988, Kissimmee, FL, pp. 1525-1529.

32. Karwatzki, J.M., "Expert Systems in Engineering,"Computer-Aided Engineering Journal, April, 1987, pp. 99-102.

33. Kimura, F., and H. Suzuki, "Variational Product Design by Constraint Propagation and Satisfaction in Product Modelling," Annals of the CIRP, Vol. 35(1), 1986, pp.75-78.

34. Kinoglu, F., D. Riley, and M. Donath, "Expert System Model of the Design Process," Conf. Proceedings of AutoFact '85, Nov. 4-7, 1985, Detroit, MI, pp. 7/14-27.

35. King, R.I., and J.G. McDonald, "Product Design Implications of New High Speed Milling Techniques," in Milling Methods and Machines, B.K. Lambert, editor, SME, Dearborn, MI.

36. Kitzmiller, C.T., and J.S. Kowalik, "Coupling Symbolic and Numeric Computing in Knowledge-Based Systems," AI Magazine, Summer, 1987, pp. 85-90.

37. Komanduri, R., J. McGee, R.A. Thompson, J.P. Covey, F.J. Tipnis, R.M. Stach, and R.I. King, "On a Methodology for Establishing the Machine Tool System Requirements for High-Speed Machining," High Speed Machining Proceedings of the WAM/ASME, New Orleans, LA, Dec 9-14, 1984, pp. 37-68.

38. Korane, K.J., "Applying Expert Systems to Mechanical Design," Machine Design, Dec. 11, 1986, pp. 121-124.

39. Kraus, J., J.J. Blech, and S.G. Braun, "In Situ Determination of Rolling Bearing Stiffness and Damping by Modal Analysis," Journal of Vibration, Acoustics, Stress, and Reliability in Design, Trans. ASME, Vol. 109, July, 1987, pp. 235-240.

40. LoPiccolo, P.J., "Expert-System Shells," Engineering Tools, May, 1988, pp. 64-71.

41. Lu, S.C-Y., "Applications of Knowledge-Based Expert Systems in Automated Manufacturing," Proceedings of the 1985 ASME International Computers in Engineering Conf., Aug. 4-8, Boston, MA, 1985, pp. 425-431.

42. Markus, A., and J. Hatvany, "Matching AI Tools to Engineering Requirements," Annals of the CIRP, Vol. 36(1), 1987, pp. 311-315.

43. Meirovitch, L., editor, Computational Methods in Structural Dynamics, Sijthof & Noordhoff, Rockville, MD, 1980.

44. Milacic, V.R., "How to Build Expert System," Annals of the CIRP, Vol. 35(2), 1986, pp. 445-450.

45. Opitz, H., R. Noppen, "Evaluation of Computer-Aided Methods for the Design of Machine Tool Structures," Annals of the CIRP, Vol. 22(2), 1973, pp. 227-231.

46. Orelup, M.F., and J.R. Dixon, "Dominic II: More Progress Toward Domain Independent Design by Iterative Redesign," Proceedings of the WAM of ASME, Boston, MA, Dec 13-18, 1987, pp. 67-80.

47. Paz, Mario, Structural Dynamics, Van Nostran Reinhold Co., New York, 1985.

48. Pittroff, H., and U.A. Rimrott, "Stiffness of Machine Tool Spindles," ASME, 77-WAM Proceedings, 1977.

49. Press, W.H., B. Flannery, S. Teukolsky, and W. Vetterling, Numerical Recipes, Cambridge University Press, New York, 1986.

50. Pruvot, F.C., "High Speed Bearings for Machine Tool Spindles," Annals of the CIRP, Vol. 29(1), 1980, pp. 293-297.

51. Rabins, M., S. Fenves, A. Seireg, G. Nadler, H. Richarson, H. Clark, "Design Theory and Methodology - A New Discipline," Mechanical Engineering, August, 1986, pp. 23-28.

52. Rajaram, N.S., "Expert Systems: The Cutting Edge of Artificial Intelligence," Robotics Engineering, May, 1986, pp. 16-20.

53. Reddy, V.R., and A.M. Sharan, "The Finite Element Modelled Design of Lathe Spindles: The Static and Dynamic Analysis," Journal of Vibration, Acoustics, Stress, and Reliability in Design, Trans. ASME, Vol. 109, Oct, 1987, pp. 407-415.

54. Rouse, N.E., "Design Optimization Goes Commercial," Machine Design, Oct 23, 1986, pp. 84-87.

55. Rouse, N.E., "More Smarts for Desktop Computers," Machine Design, Sept. 11, 1986, pp.69-70.

56. Rulemaster 2, A Reference Manual, Radian Corporation, Austin, TX, Sept., 1987.

57. Sandor, G.N., "The Seven Stages of Engineering Design," Mechanical Engineering, April, 1964, pp. 21-25.

58. Schank, R.C., "What is AI, Anyway?" AI Magazine, Winter, 1987, pp. 54-65.

59. Setco, "Boring and Milling Spindles," Catalog #181B, Cincinnati, OH, 1986.

60. Singhvi, S.C., C. Balasubrahmanyam, N.K. Seth, and A.K. Gangopaphyay, "Functional Optimization of Spindle Bearing Systems," Annals of the CIRP, Vol. 29(1), 1980, pp. 263-268.

61. SKF, Engineering Data Catalog, SKF Industries, Inc., Philadelphia, PA., 1980.

62. Smith, S., and J. Tlusty, "Update on High-Speed Milling Dynamics," Symposium on Intelligent and Integrated Analysis and Synthesis, ASME, WAM, PED Vol. 25, 1987.

63. Sohlenius, G., and T. Kjellberg, "Artificial Intelligence and Its Potential in the Manufacturing System," Annals of the CIRP, Vol. 35(2), 1986, pp. 425-432.

64. Sommerville, I., Software Engineering, Addison-Wesley Publishing Co., London, England, 1985.

65. Steele, J.M., "Guidelines for Dynamic Modeling," CAE, Jan., 1987, pp. 58-60.

66. Stern, E.L., "Structural Modeling and Modification in Generalized Modal Coordinates," Internal Report, University of Florida, Gainesville, FL, Dec., 1987.

67. Stevens, W.P., Using Structured Design, John Wiley & Sons, New York, 1981.

68. Stone, B.J., "The State of the Art in the Measurement of the Stiffness and Damping of Rolling Element Bearings," Annals of the CIRP, Vol. 31(2), 1982, 1982.

69. Tlusty, J., "Machine Dynamics," Chapter 3, Handbook of High-Speed Machining Technology, R.I. King, editor, Chapman and Hall, New York, 1985.

70. Tlusty, J., and F. Ismail, "Special Aspects of Chatter in Milling," Journal of Vibration, Acousitics, Stress, and Reliability in Design, Trans. ASME, Vol. 105, Jan., 1983, pp. 24-32.

71. Tlusty, J., S. Smith, I. Hernandez, and Y. Tarng, "High Speed High Power Spindles with Roller Bearings," Annals of the CIRP, Vol. 36(1), 1987, pp. 267-272.

72. Tlusty, J., R. Livingston IV, and Y.B. Teng, "Nonlinearities in Spindle Bearings and Their Effects," Annals of the CIRP, Vol. 35(1), 1986, pp. 269-273.

73. Ulrich, K., and W. Seering, "Conceptual Design: Synthesis of Systems of Components," Proceedings of the WAM of ASME, Boston, MA, Dec 13-18, 1987, pp. 57-66.

74. Vanderplaats, G.N., "Computational Requirements and Trends in Design Optimization," Cray Channels, Winter, 1987, pp. 16-21.

75. Vanderplaats, G.N., Numerical Optimization Techniques for Engineering Design with Application, McGraw-Hill, New York, 1984.

76. Villers, P., "Computers for Conceptual Design," CAE, May, 1986, pp. 67-69.

77. Voelcker, H.B., "Modeling in the Design Process," in Design and Analysis of Integrated Manufacturing Systems, W. Dale Compton, editor, National Academy Press, New York, 1988, pp. 167-199.

78. Waleckx, J., "Best Compromise Stiffness/Speed for Precision Machine Tool Spindles," Ball Bearing Journal No. 208, Aug, 1981, pp. 11-25.

79. Weck, M., Handbook of Machine Tools, Vol. 2, Wiley Heyden, Chichester, England, 1984.

80. Winston, P.H., Artificial Intelligence, Addison-Wesley Publishing Co., Reading, MA, 1984.

81. Yourdon, E., and L.L. Constantine, Structured Design, Prentice-Hall, Inc., Englewood Cliffs, NJ, 1979.

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This report is dedicated to the author's friends and family, with special thanks to all those who provided guidance and support at critical times. The author additionally wishes to thank Shari Gnat for her motivating support and understanding of student's hours.

I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.

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